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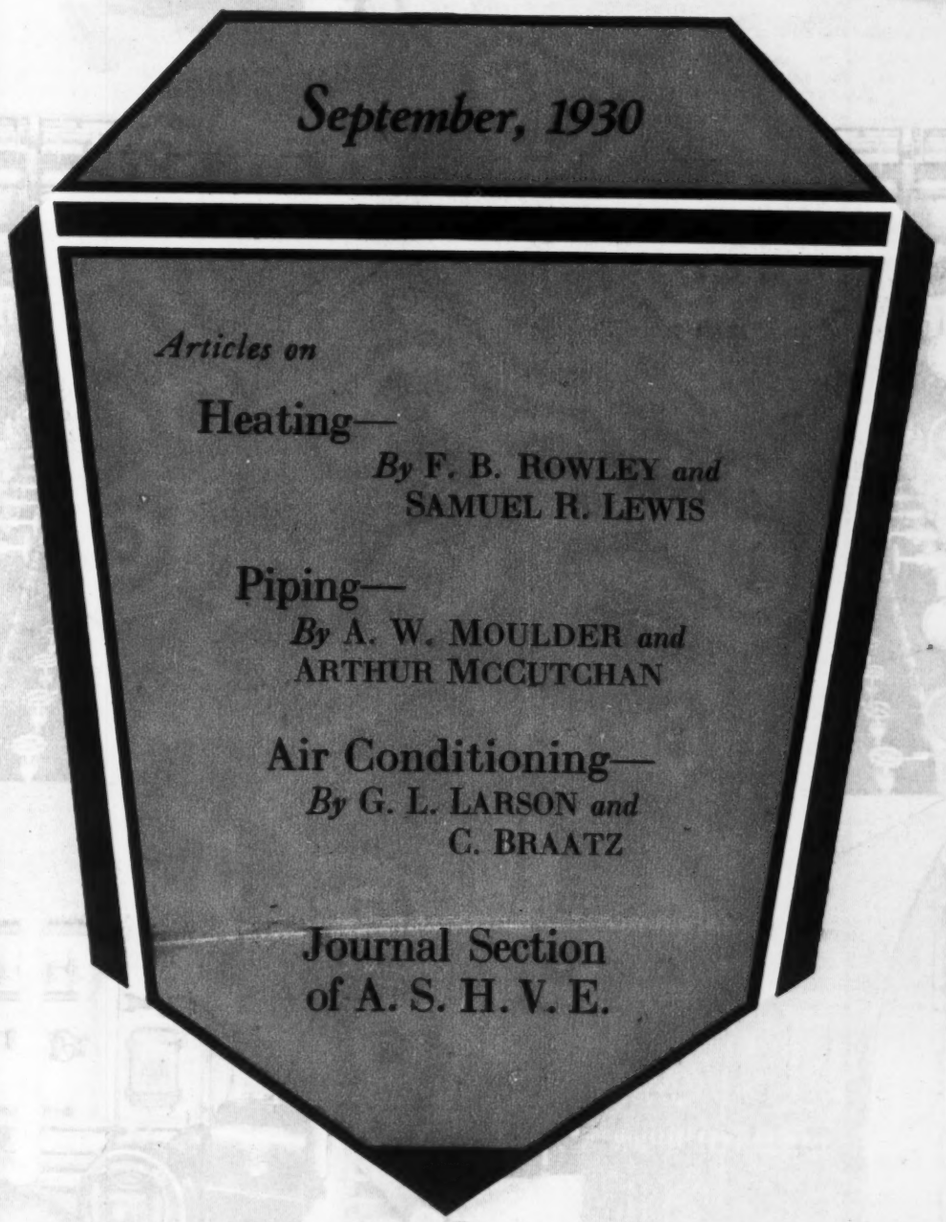
September, 1930
Articles on
Heating—
By F. B. ROWLEY and SAMUEL R. LEWIS
Piping—
By A. W. MOULDER and ARTHUR McCUTCHAN
Air Conditioning—
By G. L. LARSON and C. BRAATZ
Journal Section of A. S. H. V. E.

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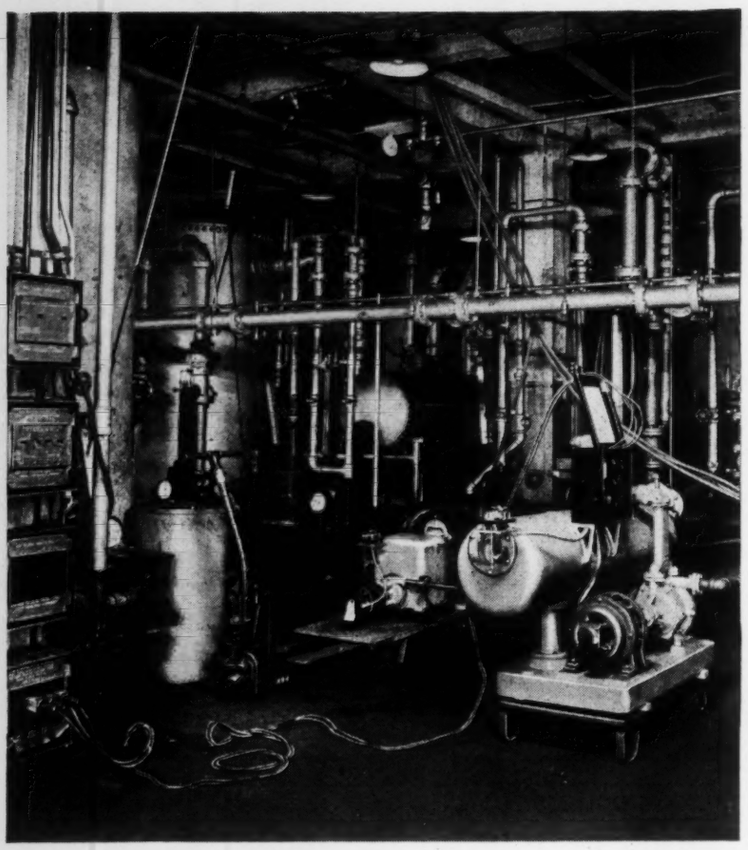

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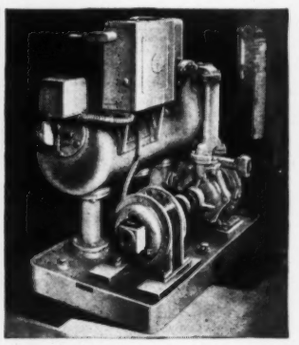

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Heating · Piping and Air Conditioning

Vol. II | September, 1930 | No. 9

Peak Load *must be considered in* Design of Air Conditioning Systems

By G. L. Larson[1] and C. Braatz[2]

IN STATES where there are well defined codes on ventilation, there is too much of a tendency for designers to blindly comply with the *minimum* requirements of the codes, giving little or no thought to the *maximum* requirements that the ventilating system will be called upon to meet from time to time.

Even the best thought out ventilation code can only be a code of *minimum requirements*. It cannot possibly set forth definite rules to cover every unknown factor and every peak load condition that may arise. At its best, a code can only set down certain minimum limits and therefore the success of any ventilating system must depend upon the foresight and experience of the ventilating engineer. The failure of a ventilating system cannot be excused by the overworked statement that "it meets the requirements of the state code." The fact must not be overlooked that the average state ventilating code is a code of *minimum requirements*. Blindly using these minimum values in connection with the design of ventilating systems for all varieties of rooms may or may not result in successful application, depending upon the number and type of peak load conditions that may be encountered.

Anticipating the ventilation demands of the average school class room is a simple matter, because the conditions of occupancy are practically constant throughout the entire school year and in such a case the application of minimum code requirements gives satisfactory results. Also in other types of buildings where the floor space per occupant is practically constant and relatively large, the problem of maintaining a reasonably comfortable atmosphere is not particularly difficult and is easy to anticipate in design.

However, many buildings contain rooms which are called upon to accommodate large crowds at intervals causing abnormal conditions to arise which are hard to anticipate in advance. Even if they are anticipated, these conditions may not be frequent enough to justify the installation of elaborate air conditioning equipment or, even though justified, may have to be omitted by virtue of limited funds. The minimum requirements of a code may be sufficient to cover the average conditions of occupancy

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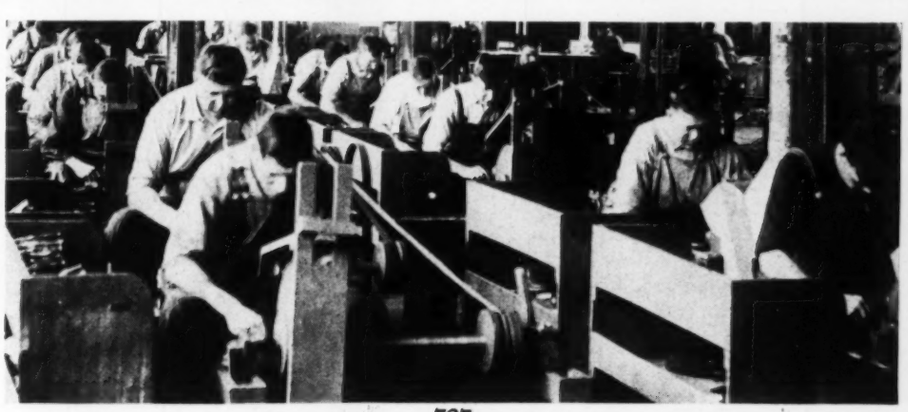


of such a room but entirely inadequate for abnormal peak conditions.

Hall Used as Lounging Room

It is the purpose of this article to analyze very briefly the abnormal conditions which may arise in a room that would be considered well ventilated under normal conditions.

The room that we will use for this purpose is the Great Hall in the new Memorial Union building at the University of Wisconsin. This room contains 5040 square feet of floor space and the ventilating system was designed to supply 2 cubic feet of air per square foot of floor. The rpm of the fan was later increased to deliver 12,300 c.f.m. to the room, approximately 2½ cubic feet of air per square foot of floor space. The state ventilating code requires that the minimum air supply to be provided and maintained for occupied areas of this class shall be not less than 1½ cubic feet per minute per square foot of the floor area served.

Under normal conditions of occupancy, the Great Hall is used as a lounging room by the students. Friday nights, there is a student dance held there attended by 100 or 150 couples at the most and occasionally, the room is used for banquets. No particular difficulty is experienced in ventilating the room under these conditions.

Dances Crowd Hall

But on nights such as the annual junior prom or the annual military ball, the story is quite different. These events result in such crowded conditions that proper ventilation is almost impossible. On such nights, the room becomes an interesting laboratory for the air conditioning engineer.

On the night of the 1930 junior prom dance, there were installed in the Great Hall several recording thermometers and a humidity recorder by means of which a history of temperature and humidity conditions in the Hall was obtained. The results appear in Fig. 1, together with outside weather conditions, data for which was furnished by the *U. S. Weather Bureau*. The chart should be read from right to left.

Recording thermometers were placed in one of the fresh air supply grilles to the room and in one of the vent openings. The humidity recorder measured the humidity of the air passing out through one of the vent openings. This particular instrument is very sensitive. A study of Fig. 1 will show that it recorded the activity of each dance very clearly.

Room Cooled Before Arrival of Crowd

Previous experience indicated that it would be necessary to cool the building before the crowd arrived in order that the structure itself might absorb some of the heat developed by the crowd. Consequently, all heat was turned off early in the afternoon and the ventilating system was made to discharge cold air into the room for several hours before the dance started. The effect on the room temperature is shown by the line marked "Outgoing Air."

It is quite common information that the temperature recorded by an ordinary dry bulb thermometer alone is but a relative indication of a person's feeling of comfort or discomfort; because from experience, we know that we may have a greater feeling of warmth at a temperature of 68 F, for example, than at 72 F if the moisture content of the air in the latter case is considerably lower than in the former. Experiments have shown[3] that the majority of people feel equally comfortable over quite a wide range of dry bulb temperatures providing the moisture content of the air, or in other words its relative humidity, is varied to suit the particular temperature under consideration. Combinations of temperature and relative humidity for which the same degree of comfort is experienced have been determined for various conditions of air movement and plotted as *equal comfort lines,* or, as they are sometimes called, *effective temperature lines,* with respect to relative humidity, and wet and dry bulb temperature.[3] In still air, it was found that 97 per cent of people are most comfortable at an effective temperature of 64, which, on the dry bulb temperature scale, includes the range between 64 and 72 F. For a dry bulb temperature of 68, this corresponds to a relative humidity of about 40 per cent.

Moisture Content of Air Important

Under normal conditions of operation, the problem of maintaining a condition of comfort in most buildings is not so much one of supplying sufficient heat, (assuming of course, a properly designed heating system of ample capacity) but rather one of maintaining the air in a condition such that the heat supplied will be the most effective. Ordinarily, this will require the evaporation of a considerable amount of moisture in the average building, because when the cold outside air, that replaces the warm inside air from one to three times each hour by infiltration, undergoes a temperature change from —10 to 70 F, for example, its relative humidity is reduced to such an extent that we very often live in an atmosphere that is as dry as the air over the Sahara desert.

This is accounted for by the fact that while the evaporation of snow and ice (or should we say sublimation?) on a cold day takes place very slowly, the amount of moisture required to saturate the cold air is comparatively small and hence its *relative* humidity may be very high, as indicated by Fig. 1. When this air is heated to 70 F, its capacity for moisture is very appreciably increased, and therefore its relative humidity will be very much less than it was initially. For example, air at zero F, and a relative humidity of 80 per cent, when heated to 68 F will have a relative humidity of only about 5 per cent.

Crowd Supplies too Much Moisture

In Great Hall, however, on such occasions as the junior prom and the military ball, the effect of the crowd on inside weather conditions alters the situation to such an extent that instead of the air being too dry, it may be too moist for comfort.

Two conditions arise as a result of the crowd, both

[3] 1930 Guide of the *American Society of Heating and Ventilating Engineers.*

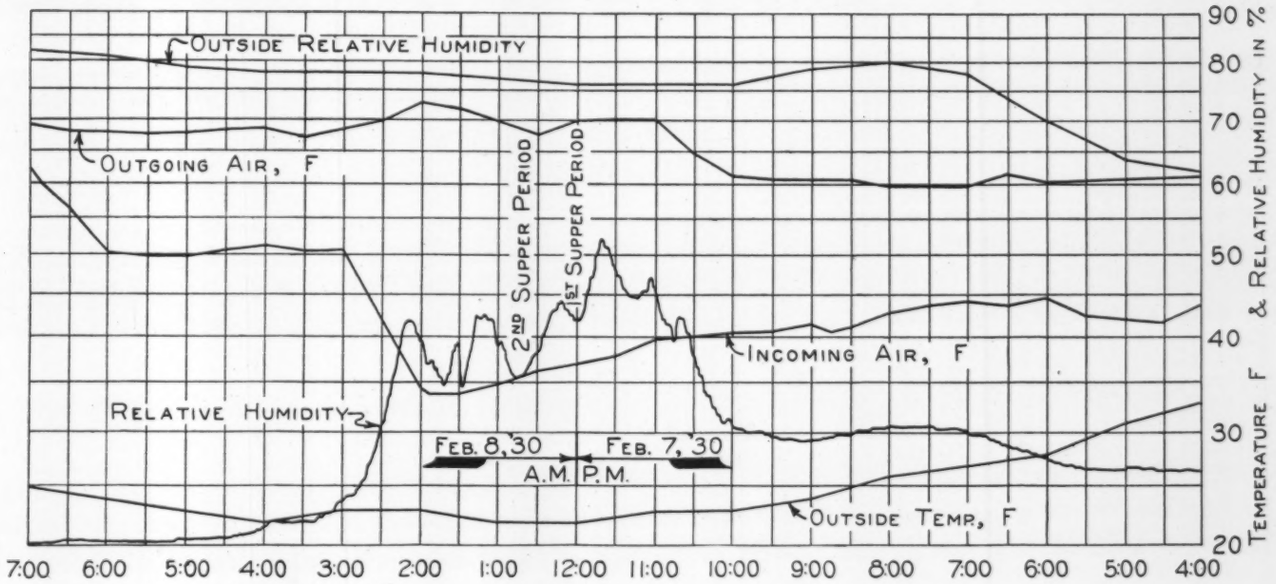


FIG. 1—TEMPERATURE AND HUMIDITY CHART FOR THE ANNUAL PROM AT THE UNIVERSITY OF WISCONSIN

of which tend toward its discomfort. Research work carried out by F. C. Houghten [4] indicates that an average individual normally clothed and at rest gives off about 400 Btu per hour, while one at work will produce between 550 and 600 Btu per hour by radiation, convection, evaporation and exhalation.

Also, the total amount of heat produced by an individual for a given degree of activeness remains practically constant, although the amount produced by any one of the foregoing methods varies. That is to say, in order to carry away this constant amount of heat produced by the body, Nature has evidently equipped us with a temperature control system by means of which she endeavors to keep the body temperature constant by maintaining equilibrium between the heat produced and the heat dissipated, by variations in the rate of evaporation, radiation and convection. For example, as the temperature of the air approaches that of the body, the heat dissipated by radiation and convection decreases, while to counteract this condition more perspiration is made available, resulting in greater evaporation. Similarly, a low relative humidity is conducive to a high rate of evaporation. Hence it is apparent that conditions may arise, such as a combination of high dry bulb temperature and high relative humidity, for which Nature's control system ceases to function properly, and we experience the familiar muggy feeling.

Heat Output of the Occupants

There may be some difference of opinion as to whether or not dancing under the crowded conditions of a prom or military ball comes under the category of work, but in the absence of data relative to what a normal strait-jacketed male and his girl-friend *can* do to the tune of "Turn on the Heat," it seems reasonable to assume 550 Btu per hour as an average figure for the evening.

On the basis of the number of tickets sold and given away, it is estimated that the attendance at the 1930 prom was about 750 couples. Assuming that two-thirds of these were in Great Hall, for the first dance at least, (this allows a space of 3½ feet square for each couple) would mean that the dancers alone were the equivalent of a heating plant having a capacity of 550,000 Btu per hour, which, for the temperature conditions that prevailed on the night of February 7-8, is approximately three times the amount of heat required to maintain the hall at a temperature comfortable for dancing, say 65 F, assuming the proper relative humidity. And had it been an extremely cold night, twenty degrees below zero let us say, the dancers would still have developed 48 per cent more than the required amount of heat. Hence, the problem of keeping the dancers comfortable, even in extreme weather, is one of cooling rather than one of heating.

Crowd Acts as Humidifier

In addition to substituting for the heating plant of the building, the crowd also acts as a humidifier, which, *to a certain extent,* is desirable for reasons already mentioned. However, if the amount and condition of the air entering the building is not such as to provide for carrying away the moisture dissipated by the crowd before the relative humidity becomes too high, conditions of comfort are still further aggravated. Hence, to counteract the tendency of the dancers to over-heat and over-humidify the air in the hall, air must be supplied at a lower temperature or in larger amounts, or the condition met by combinations of the two methods. The minimum temperature at which the air can be supplied is limited by the actual outdoor temperature or by the drafts created if the air is too cold when it enters the room. Therefore, to prevent the relative humidity from becoming too high, larger amounts of air must be depended upon in the absence of dehumidifying equipment.

The heating and humidifying effect of the dancers is clearly shown by Fig. 1. The inside relative

[4] Director, Research Laboratory, *American Society of Heating and Ventilating Engineers.*

humidity curve indicates that the grand march started about 9:45, at which time the relative humidity began to climb slowly for about twenty minutes, then to drop off momentarily during the picture taking ceremony and speech by the prom chairman, which was followed by a very decided rise during the first dance. From here on each of the following dances has a decided hump of its own on the relative humidity curve.

Note that the temperature of the air leaving the hall rose from about 61 to 70 F during the first dance in spite of the fact that 12,300 cubic feet of 40 deg. air was being discharged into the room every minute. Note also that it was necessary to gradually decrease the temperature of the incoming air throughout the entire evening in order to maintain the room at a comfortable temperature. Inasmuch as the average relative humidity between 10:00 P. M. and 2:00 A. M. was about 40 per cent, the chart would imply that one should have been reasonably comfortable at the 1930 prom, except perhaps during a short period near midnight when the relative humidity rose to about 53 per cent.

Air Supply

In the Great Hall, fresh air is supplied through grilles in the dome of the ceiling, after passing through heating coils, if necessary, and is exhausted through grilles in the south wall benches. Obviously, as long as the temperature of the outside air is sufficiently low, air can be supplied at a temperature such that in falling from the dome to the breathing line, it will be tempered to a value conducive to comfort. When the outside temperature is 22 F, the calculated heat loss of the Great Hall is 188,000 Btu per hour with an inside temperature of 65 F. Since the heat developed by the dancers is 550,000 Btu, there is an excess of 550,000 — 188,000 = 362,000 Btu per hour that must be carried away by the ventilating system if the room is to be maintained at 65 F. To calculate the temperature at which fresh air must be supplied in order to carry away the excess heat developed by dancers, the following relationship can be used:

$$(1) \qquad H = \frac{Q \times 60 \times (T_2 - T_1)}{55}$$

where H = Excess Btu per hour = 362,000 Btu.
Q = Cubic feet of air supplied per minute = 12,300.
T_1 = Temperature of incoming air.
T_2 = Final temperature = 65 F.
55 = Number of cubic feet of air that can be raised in temperature 1 F by the addition of one Btu.

$$(2) \text{ or } \qquad T_1 = T_2 - \frac{55H}{Q \times 60} = 38 \text{ F}$$

The actual average values of T_1 and T_2 for the night of Feb. 7-8 from Fig. 1, was 37 F and 67 F, respectively.

Let us assume now that the outside temperature is 40 F instead of the actual 22 F, and let us further assume that the heat loss from the room is directly proportional to the difference between the inside and outside temperatures. The heat supplied by the dancers is still 550,000 Btu per hour but the heat loss from the room is reduced to 109,000 Btu per hour and the excess heat to be carried away now becomes 441,000 Btu instead of 362,000 Btu per hour. Solving again for T_1 from equation (2):

$$T_1 = 65 - \frac{55 \times 441{,}000}{12{,}300 \times 60} = 32 \text{ F}$$

which means that to maintain an inside temperature of 65 F, would necessitate the installation of some sort of cooling system, or increasing the supply of air, Q, from 12,300 c.f.m. to 16,200 c.f.m. obtained by solving for Q in equation (1):

$$Q = \frac{55 \times 441{,}000}{60 \times (65 - 40)} = 16{,}200$$

It is true that opening the windows on such occasions will relieve conditions for dancing, but with the outside temperature at 40 F, such a procedure is hardly advisable from the standpoint of uncomfortable drafts for those who sit along the walls during rest periods.

Experimental data are also available[5] by means of which the approximate condition of the air with respect to relative humidity could also be predetermined for the purpose of designing air conditioning equipment, but for the purpose of this article the actual average amount of moisture evaporated by the dancers per hour can be readily calculated from the information given on Fig. 1, and knowing the amount of air supplied to the hall per unit time by the fan system.

From a psychrometric chart, (see *A. S. H. V. E. Guide*) it is found that the moisture content of air at 22 F, and a relative humidity of 77 per cent, (using average values for the night of February 7-8) is 12 grains per pound, while at 67 F, and a relative humidity of 41 per cent the moisture content is 47

[5] 1930 Guide, *A. S. H. V. E.*, page 100.

grains per pound. Hence with the fan system supplying 12,300 c.f.m., the average rate of moisture evaporation by the crowd must have been about 276 pounds or 33 gallons per hour. Near midnight, at which time Fig. 1 indicates the crowd was at its best, the rate of evaporation went up to about 43 gallons per hour. Fig. 1 also clearly shows that there were five dances following the grand march which apparently started about 9:45 P. M. With 500 couples on the floor, the latter figure would mean an evaporation of about one-fifth of a pint or 0.2 of a pound per capita per dance; or a loss in weight by evaporation, for the entire evening, of one pound per person.

The foregoing illustrates the necessity of making a complete analysis of probable conditions in designing the ventilating equipment for certain types of rooms. The minimum requirement of the state ventilating code provides for less than half enough air to properly ventilate such a room as the Great Hall under the peak load conditions that arise from time to time.

Air for Office Building Cleaned and Dehumidified

The entire building of the R. J. Reynolds Tobacco Company at Winston-Salem, North Carolina, with the exception of portions of the main entrance corridor is heated by direct radiation. Steam for heat is obtained from the company's power plant and enters the building at 100 lb., is reduced to 15 lb. for ventilation stacks and kitchen equipment, and to 3 lb. for the direct heating system. Radiators, except in a few locations, are of the cast iron, wall hung type. In some places brass radiators have been used. All direct radiators throughout are equipped with thermostatic traps on the return end, and on the supply with temperature regulating valves.

Air is supplied and exhausted from the following parts of the building by full-housed centrifugal fans with single inlets and outlets.

Sub-Basement Supply Fan

Located in sub-basement. Supplies 5,784 cubic feet to the sub-basement machinery room, 3,300 cubic feet to the employment department waiting room in the basement, and 2,000 cubic feet to the main entrance lobby which has a booster heater in the tunnel just under the lobby.

General Supply Fan

Located in sub-basement. Supplies 7,958 cubic feet to the employment department, pay roll department, inside storage rooms and barber shop, all located in the basement.

Basement Cafeteria Supply Fan

Located in sub-basement. Supplies 23,327 cubic feet to the basement cafeteria which includes toilets, lobby, dining rooms, kitchen and bakery.

10th Floor Supply Fan

Located in tower. Supplies 5,290 cubic feet to the cafeteria located on the 10th floor and also to a private dining room on this floor.

19th Floor Supply Fan

Located in the tower. Supplies 1,490 cubic feet to inside rooms on the 19th floor.

General Exhaust Fan

Located in sub-basement. Exhausts 11,422 cubic feet from the sub-basement machine room, engineers' toilet, employment department waiting room in basement, and to inside offices in the mezzanine floor.

Toilet Exhaust Fan

Located in the tower. Exhausts 12,400 cubic feet from toilets and slop sinks on all floors.

Basement Cafeteria Exhaust Fan

Located in the sub-basement. Exhausts 18,147 cubic feet from the basement cafeteria, which includes toilets, coat room, dining rooms, kitchen, bakery and storage rooms.

Basement Range Hood Exhaust Fan

Located in the tunnel under the main lobby. Exhausts 4,200 cubic feet from the range hood over the basement kitchen range.

10th Floor Range Hood Exhaust Fan

Located in the tower fan room. Exhausts 1,500 cubic feet from the range hood in 10th floor kitchen. (Note: This fan is not in use, as all cooking is done in basement kitchen and food carried up on dumbwaiter to 10th floor).

10th Floor Exhaust Fan

Located in tower fan room. Exhausts 3,970 cubic feet from over the steam tables, coffee urns and kitchen of the 10th floor cafeteria.

The main fresh air intake is at the second floor level and supplies the sub-basement, general supply and basement cafeteria fans and the 10th floor supply fans. The 19th floor supply fans take fresh air from the tower.

The general exhaust fan discharges on to the ninth floor roof and all others discharge on to the tower roof.

The supply fans are equipped with air filters and dehumidifiers composed of baffle plates, spray heads and eliminator plates.

Temperature regulation, designed for summer and winter use, is used on the supply fans which control the temperature at the fresh air intakes, air washers and also the preheaters and reheaters.

In addition to this temperature control there are wall mounted positive action thermostats in the main entrance and basement and 10th floor cafeterias and the employment waiting room in the basement.

Temperature Distribution in Field House at University of Minnesota

By F. B. Rowley*

HEATING and ventilating a large open space, such as the University of Minnesota field house, presents entirely different problems from those met with in average low-ceilinged buildings or even with the average high-ceilinged building such as a theater, which has but very little heat loss to the outside. For the average theater, the major problem is that of purifying and cooling the air, as heat supplied by the audience is usually in excess of the heat losses from the building.

FIG. 1—UNIVERSITY OF MINNESOTA FIELD HOUSE

In the field house, which has large exposed outside areas and, necessarily, high heat losses, the problem is one of both heating and ventilating. During the cold winter season, when the building has the greatest amount of use and when the outside temperature may often range from 0 to 25 below zero, one of the major problems is to supply the proper amount of heat to the building and to distribute this heat where needed. At other times, when the outside temperatures are higher or when there has been a large crowd in the building for some time, the problem is one of ventilation and proper temperature control throughout the building. So far as the heating problem is concerned, it is necessary to supply enough heat to take care of the heat losses from the building and to supply the heat in such a manner as to give the proper temperature distribution over the different parts of the building. The ventilation problem is, first, one of comfort for spectators and players, and, second, one of humidity control to prevent a fog forming over the playing floor and possible condensation on the roof. There is also the question of air change, but this is not a difficult problem for a building of such large volume and one that is located in a section where the outside air may be taken in with no special treatment.

Problem Simplified by Stratification

As a matter of comfort, it is desirable to keep the playing floor at a temperature not above 60 F, while the balconies for the audience should be maintained at an average of 68 F. This part of the problem is simplified by the fact that the balconies, being above the playing floor, are in a higher temperature zone, providing the air has a tendency to stratify at all. It is, however, a problem to keep the air at the same temperature over all of the seats, as those at the top of the second balcony are 33 feet higher than the lower seats of the first balcony.

Humidity

The maximum humidity possible without causing trouble depends upon the temperature near the floor line where a fog is most likely to appear and the temperature of the air on the under-side of the roof. Some previous designers of buildings of this nature have found trouble along these lines. The natural remedies are, first, to insulate the roof, and, second, to provide for sufficient outside air to be drawn in to the building through the heaters to keep the relative humidity down below the saturation point.

The Building

The main floor of the field house, with which we are concerned, is 232 feet wide and 440 feet long. The distance from the playing floor to the roof is 40 feet along the side walls and 100 feet at the center section. The side and end walls are constructed of brick and the roof with a steel deck, using 1 in. of insulating material.

Fig. 1 shows the exterior of the building. Fig. 2 is a plan view, giving the location of the various heaters used, and Fig. 3 is a cross-sectional view showing the elevation of the heaters and the ducts used for air distribution. Fig. 3 shows, also, the balconies along the side walls and the points at which air temperatures were taken for the various tests.

The main part of the building is heated by indirect fan heaters; twelve of the largest of these heaters are placed along the side walls just below the upper balcony, six along each side as indicated in Figs. 2 and 3. Each large unit is heated by a total of 954 feet of fin-type radiation. The air is driven through the heater by a fan capable of supplying 10,500 cubic

* Director of Experimental Engrg. Laboratory, University of Minnesota.

feet per minute, with an outlet velocity from the discharge duct of 1,500 feet per minute. As shown in the elevation view of Fig. 3, louvers are provided in the intake side of the fan so that the air can be either re-circulated in the building or outside air can be taken in and heated as desired.

In addition to the large heaters placed beneath the second balcony, there are five small heaters placed on each side near the floor under the first balcony. Each of these heaters has an intake box to draw the air from the floor line and deliver it to the heater about 6 feet above the floor. The heating section for each of these small heaters consists of a single section fin-heater 29 in. x 32 in. overall, combined with a direct connection motor-driven propeller fan. Each fan has a capacity of 2,000 cubic feet of air per minute, and the heater a steam-condensing capacity of 150 pounds per hour with 60 F entering air.

There are two similar heaters without the intake boxes placed on each end wall about thirty feet above the floor. In each corner of the building near the floor line, there is a heater with intake box similar to those on the side walls. The capacity of the heaters is 5,000 c.f.m., with a steam-condensing capacity of 370 pounds per hour. There are two 18,000 c.f.m. exhaust fans placed in the roof equally spaced from the ends.

From the layout, it will be noted that the largest proportion of heat was supplied by the upper heaters. The air for these heaters, when re-circulated air was used, was taken from a point 36 feet above the floor and the major part of it was delivered through a 28 in. x 32 in. duct 52 feet from the floor, directed downward at approximately 45 degrees. A small portion of the air was delivered through a 12 in. x 22 in. duct at a point 84 feet above the floor, close to the roof. The outlet velocity from the heating ducts was designed for approximately 1,500 feet per minute.

Control of Temperatures

As automatic heat control was not installed, temperatures were controlled manually by one of the following methods:

1. By shutting off the entire system of fans and heaters.
2. By shutting off a part or all of the fans, leaving the steam turned on.
3. By shutting off a part or all of the heaters, leaving the fans on.

There were other combinations of manual control possible, but the one which was easiest for the operator and most often used was to, first, heat the building with all equipment turned on, and then shut off a part or all of the fans, leaving the heaters on. The upper fans were usually turned off when this method was used.

With the roof insulated with one inch of insulating material, no trouble was experienced with condensation and the humidity has never reached a point high enough to give trouble with a fog over the playing floor during the basket-ball games. Temperature distribution and control were therefore the main problems for study.

Method of Making Temperature Tests

Since there are large exposed areas, the outside air temperature and wind velocities are large factors in the heating requirements. The size of the audience and length of time in the building also have their effects on the heat required. The method of operation was also a vital factor. As there were no automatic controls provided, it was left largely up

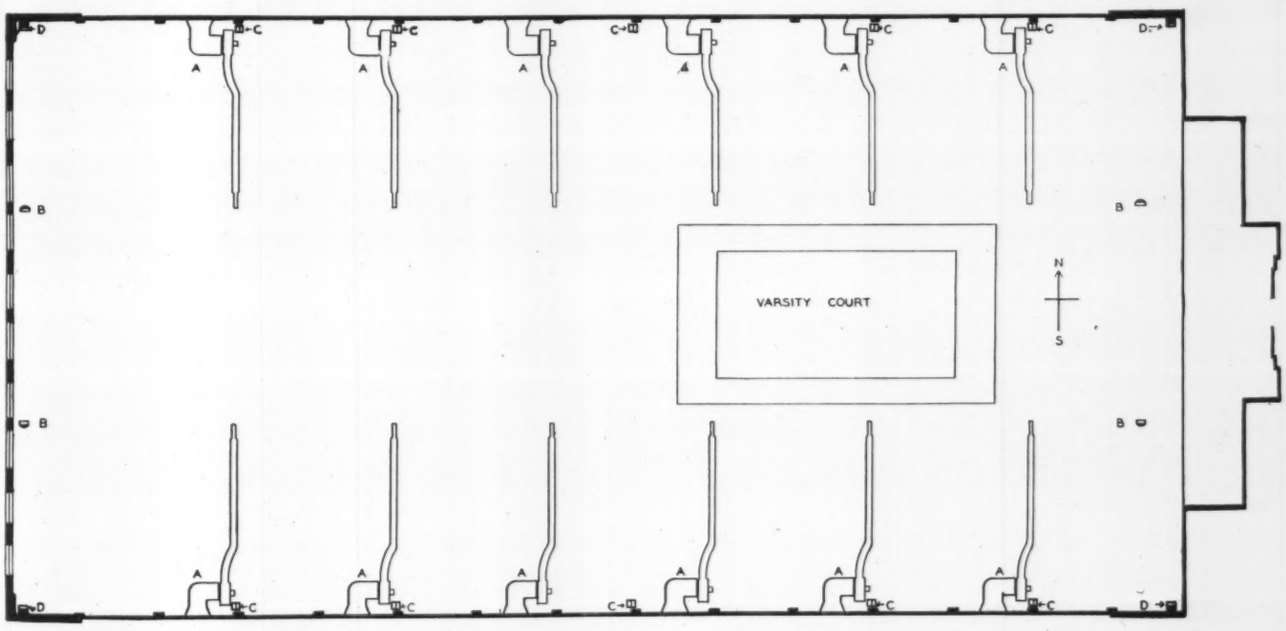


Fig. 2—Plan of Field House Showing Location of Heaters. *A*, 10,500 c.f.m. Heaters Located Under Second Balcony; *B*, 2000 c.f.m. Heaters Located at End Walls About 30 Ft. Above Floor; *C*, 2000 c.f.m. Heaters Located at Side Walls About 5 Ft. Above Floor; *D*, 5000 c.f.m. Heaters Located Near Floor in Corners of Building

to the operator as to what method should be used to control the temperatures.

In order to determine temperature distribution and other operating characteristics of the system, tests were run in the early part of 1929, covering the following conditions:

(a) All heaters and fans turned on to determine temperature distribution throughout the building.

(b) All heaters and fans running to warm up the building with large fan shut off during the test to determine temperature distribution and changes under this condition.

(c) All fans and heaters running to bring the building up to temperature before a basket-ball game with the upper fans shut off during the game.

(d) All heaters and fans started one and one-half hours before test, upper heater shut off, and all other equipment operating during test.

(e) All heaters shut off, large fans running to show temperature distribution and air circulation.

The air temperatures for the various tests were taken with both copper constantan thermocouples and mercury thermometers. The thermocouples were used to take the temperatures from the roof to the floor at the center line of the building. A couple with 100 foot leads was so arranged that the couple could be placed at any point between the floor and ceiling by means of a rope passing over a pulley at the ceiling and operated from the floor. The thermocouple leads were wound on a reel with the instrument connections brought out through the center of the reel. By this means, it was a simple matter to get temperature at any elevation.

Mercury thermometers were used to get the temperatures at the floor line and at the different seat elevations. Four thermometers were used for floor temperatures and twelve thermometers for the seats, three for each balcony on each side of the building. The thermometers were placed to give the cross-sectioned temperatures near the center of the building.

The floor temperatures at the four points were found to run very uniformly, the maximum variation for the four points being 2 F. In the data reported here, these temperatures have been averaged and given as the floor temperature. The same temperature uniformity was observed at the seats for corresponding elevations; therefore, the temperatures were averaged for the same elevation for the two

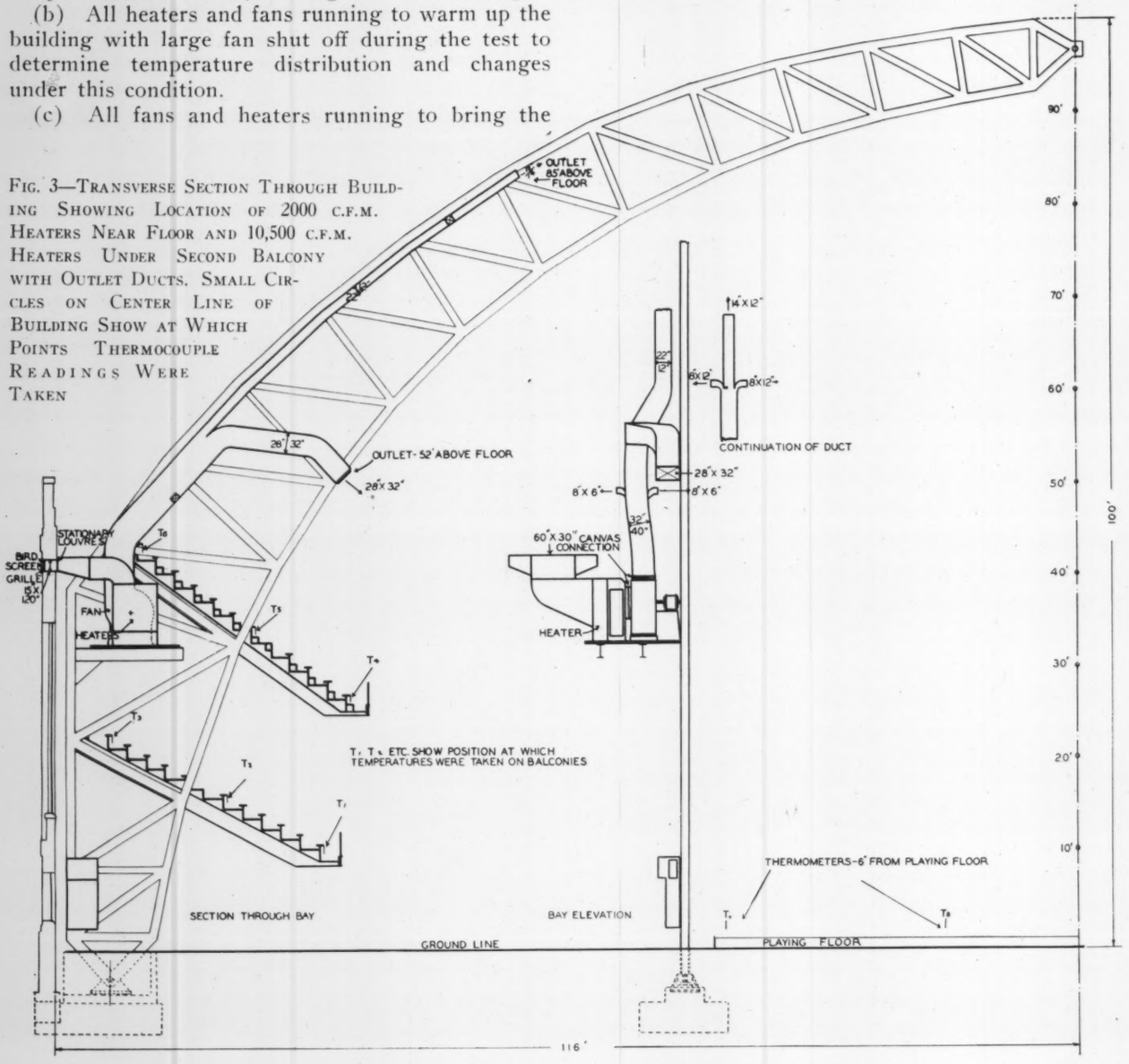


FIG. 3—TRANSVERSE SECTION THROUGH BUILDING SHOWING LOCATION OF 2000 C.F.M. HEATERS NEAR FLOOR AND 10,500 C.F.M. HEATERS UNDER SECOND BALCONY WITH OUTLET DUCTS. SMALL CIRCLES ON CENTER LINE OF BUILDING SHOW AT WHICH POINTS THERMOCOUPLE READINGS WERE TAKEN

sides of the building, with the exception of one test which was taken with the audience in the building.

In this test, the larger proportion of the audience was on the south side and the temperatures were somewhat higher on these balconies. For the vertical section through the center line of the building, temperatures were taken at 10 feet intervals. This method was satisfactory in cases where the temperatures remained constant at the various levels for a sufficient length of time to cover the range. If the temperatures were changing rapidly, it was only possible to take a few points, usually at top center and floor levels.

The temperature results obtained and the different conditions of tests are shown in Tables 1 to 6, inclusive. Table 1 shows the center line temperatures as obtained by the thermocouple, and Tables 2 to 6, show the temperatures at the seats with corresponding center line temperatures as taken during the various tests.

The temperature distribution tests were supplemented by smoke tests to show more clearly the path of the air currents. The smoke tests were made by placing smoke bombs at the intake of the heater fans and observing the path followed by the smoke from the discharge ducts of the heaters. The results of these tests are shown graphically by Figs. 4 and 5. Fig. 6 is a typical illustration of the results obtained.

Analysis of Results

The results obtained for any one operating condition can best be studied by referring to the center line temperatures in Table 1, the cross-section temperatures, Tables 2 to 6, and the smoke test for the particular condition. With all of the equipment running for 1½ hrs., the center line temperatures are shown in column 1 of Table 1. With longer periods of operation or with different outside weather conditions, these temperatures would naturally be different, but the distribution is typical for this operating condition.

The temperatures rise very rapidly from the floor to a point about 50 feet above the floor and then remain practically constant for the remainder of the distance to the roof. This condition is due to the fact that the large heaters discharge the greater part of their air at about the 50-foot line and draw it in at a point rather high at the side walls. The hot air discharged passes down a short distance, due to the exit velocity, but soon turns upward, due to its temperature. Fig. 4 shows the average results of the smoke test under these conditions.

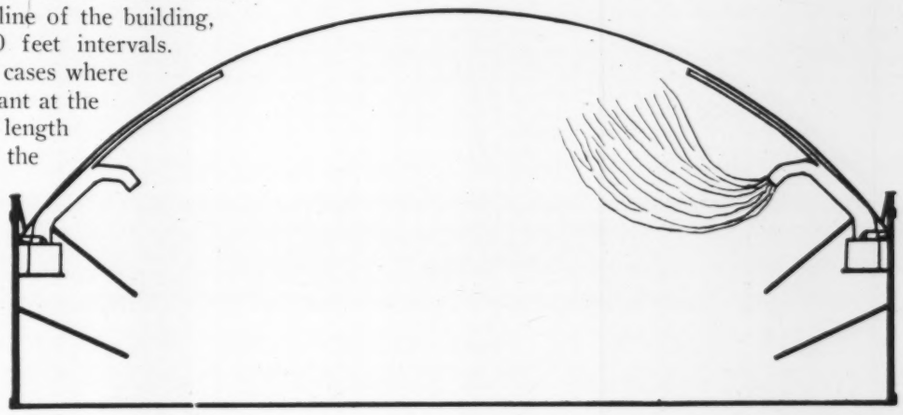


FIG. 4—DIAGRAM SHOWING TRAVEL OF SMOKE WITH HEAT TURNED ON UPPER HEATERS

Column 2 of Table 1 shows the tendency toward a more uniform temperature gradient from top to bottom after the upper fans were shut off. The temperatures for the balcony seats for this same condition are shown in Table 2. The temperatures at the seats correspond very closely to the center line temperatures of the same elevation for the same period in the test.

The maximum spread between the lower and upper balcony seats was 16.5 degrees. At this condition, however, the lower seats were too warm. As the fans were shut off, all temperatures gradually dropped, the temperature distribution became more uniform from bottom to top of building, and the temperature spread between lower and upper seats was reduced to 11.5 with the lower temperature at 66.2 degrees.

The results of smoke tests taken with all heater and fans on are shown in Fig. 4. These tests show that in heating the building, the tendency of the warm air to rise soon overcomes its downward velocity and, in this case, the greater part of the heat is supplied to the upper half of the building. The hot air being thrown out at about the 50-foot line tends to give a uniform temperature from this point to the top. As there is but very little agitation in the lower half of the building, the temperatures throughout this space assume substantially a uniform gradient from the floor to a point 50 feet above. As the balcony seats are in the lower half and mostly in the unagitated portion of the air, they also vary in temperature according to their elevation.

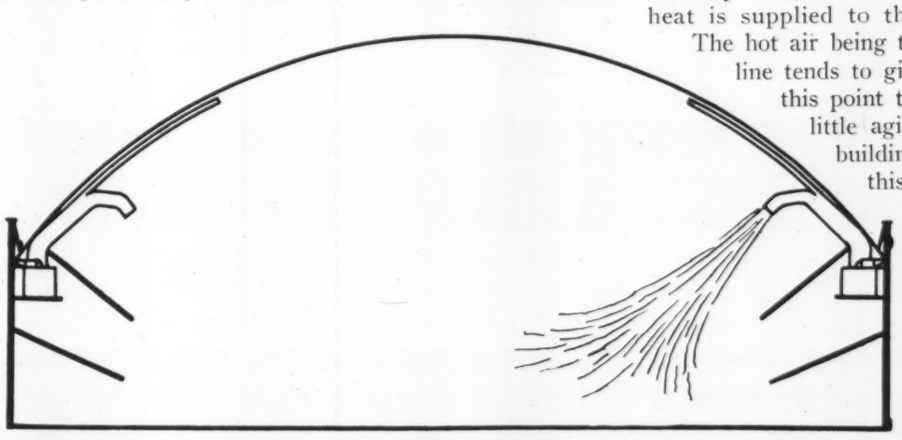


FIG. 5—DIAGRAM SHOWING TRAVEL OF SMOKE FROM HEATERS WITH HEAT SHUT OFF

As a further test to determine the conditions reached with upper fans turned off but

TABLE 1—AVERAGE TEMPERATURES, CENTER LINE OF BUILDING

FEET ABOVE FLOOR	OPERATING CONDITIONS				
	1 ALL FANS AND HEATERS RUNNING 1 HR. 30 MIN. OUTSIDE TEMP. +11° F.	2 CONTINUATION OF NO. 1. LARGE FANS OFF FOR 30 MIN.	3 ALL HEATERS ON. LOWER FANS ON. UPPER FANS OFF. OUTSIDE TEMP. —2° F.	4 ALL HEATERS OFF. LARGE FANS RUNNING FOR 1 HOUR. OUTSIDE TEMP. +24° F.	5 SAME AS 3 1-11-29
100	90	84.7	77.0	63.	72.5
90	90		77.0	63.	70.8
80	90		73.6	62.8	70.8
70	90		71.9	62.5	69.6
60	90		70.4	62.5	68.8
50	89	80.0	69.9	62.3	66.7
40	85.5		68.5	62.	65.4
30	80.5		68.9	61.8	64.5
20	75.5		66.8	61.4	62.6
10	69.8		63.0	61.	59.9
0	62	61.8	59.0	60.1	56.8

TABLE 2—ALL FANS AND HEATERS ON AT START OF TEST. LARGE FANS SHUT OFF AT 4:47. JAN. 16, '29, OUTSIDE TEMP. +11° F

TIME	TEMPERATURE AT SEATS						TEMP. BUILDING C. L.			REMARKS
	1	2	3	4	5	6	FLOOR	50 FT. ABOVE FL.	CEILING	
3:30	68.2	71.5	73.7	75.5	79.5	82.7	62		84.1	Fans running 20 min. before test.
3:45	69	72.5	74	76.5	80.0	85				
4:15	71	74.5	76	78.2	82.0	86.5				
4:45	70	73.2	76.2	78.2	82.0	86.5	62	89	90	Large fans stopped immediately after reading.
5:00	67.2	70.7	72.5	73.7	77.2	80.5				
5:15	66.2	69.5	70.5	71.7	75	77.7	61.8	80	84.7	

TABLE 3—ALL HEATERS WERE TURNED ON. LOWER FAN ON. UPPER FAN OFF FOR TEST. JAN. 25, '29, OUTSIDE TEMP. –2° F

TIME	TEMPERATURE AT SEATS						TEMP. BUILDING C. L.		
	1	2	3	4	5	6	FLOOR	50 FT. ABOVE FL.	CEILING
3:15 P. M.	65.3	66	67.8	69	70.2	70.7	60	70	75
4:00	65.2	66	67.2	68.2	69.5	69.1			
4:30	65.6	65.7	67	68	69	69	58.1	70	73

all other equipment running, the data of Table 3, and, also, column 3 of Table 1, were taken. When operating under this condition, the temperature gradient became more uniform from floor to ceiling with less of a difference between the lower and upper balcony seats. The building was heated up before this test was started by operating all of the fans and then allowed to cool down for the conditions of test.

The data of Table 4 were taken during a basketball game. The building was warmed up some time before the game and then the upper fans were shut off. Temperatures are given for both the north and south balconies, as there were about twice as many spectators on the south as on the north side.

In this test, the south balcony was slightly warmer than corresponding parts on the north balcony. The temperature dropped throughout the game, indicating that the heat loss from the building was greater than that supplied by the audience and the heating system as operated. The spread in temperature between the upper and lower balcony seats ranged from 5.1 deg. at the beginning to 8.6 deg. at the end of the test. The decrease in temperature throughout the game was greater for the lower seats than for those at the top. This was more noticable for the south balcony seats and was probably due to the larger audience on that side.

The effect on temperature distribution of shutting off all heaters and operating the fans is shown by column 4 of Table 1. After one hour of operation under this condition, there was only a three-degree temperature rise from floor to ceiling. The smoke test of Figs. 5 and 6 show that under this condition the air travels downward to the floor and the diffusion in the lower part of the building is much better. There was also some diffusion in the upper part but probably the cooling off of the upper level, by heat loss from the building, helped to lower the temperatures at the upper levels.

The effect on balcony temperature of shutting off the large heaters and allowing the fans to operate is shown in the data of Tables 5 and 6. In both cases the temperatures of the upper and lower seats are brought nearer together by operating the large fans without the heaters. Under this condition, the air discharged from the fans is diffused more nearly through the section of the building in which the balconies are located. Since it is not heated it does not rise to the ceiling. It will be noted that the evening-up is due to the dropping of the upper temperatures rather than to any change in the lower temperatures. If the outside temperatures were colder, this method of operation would not supply enough heat and it would become too cold.

These tests show that air soon seeks its temperature level. If warm air is discharged into an open space at a temperature higher than the surrounding air its course will very soon be upward regardless of the direction of discharge. If the temperature of the discharged air is lower than the surrounding air, it will continue downward until it is diffused with surrounding air. In other words, it is difficult to overcome the force of gravity on the air by outlet velocities in large open spaces. The distance to which air can be expected to travel against gravity depends upon temperature difference and the discharge velocity.

For high open spaces the problem becomes more difficult than for low confined spaces. The temperature differences from floor to ceiling may be reduced by agitation of the air, by heating with larger volumes of air with less temperature rise in the heaters, or by drawing the hot air from the top and discharging it through the heaters at the floor line. If it is desired to stratify the air, the upper levels may be easily maintained at higher temperatures by selecting the points of discharge and intakes at the upper levels. The limits of temperature controls for such

TABLE 4—DATA TAKEN DURING A BASKETBALL GAME. UPPER AND LOWER BALCONIES ON SOUTH SIDE WELL FILLED. NORTH SIDE ABOUT ½ FULL. ALL HEATERS TURNED ON. LOWER FANS RUNNING. UPPER FANS OFF EXCEPT ONE IN N. E. SECTION. FEB. 18, '29, OUTSIDE TEMP. -9°F

TIME	BALCONY TEMPERATURES										C. L. TEMP.	
	1-S	2-S	3-S	4-S	5-S	6-S	1-N	3-N	4-N	6-N	FLOOR	CEILING
8:00	71.5	74	73	75	76	76	69.8	74.2	74	75.5	61	
8:10	69	72.5	71.5	74	75.5	76	69.0	71.7	74	75.0	61	78
8:20	67	72.0	72	73.5	75.5	75.8	67.2	71.0	73.5	74	60.5	
8:30	66	71.3	71	73.0	75	75.3	66.4	70.0	72.0	73.5	60	
8:40	66	71	70.6	72.3	75	75.0	66.0	70.0	71.5	73.0	60	
8:50	66.5	71.2	70.6	72.3	75	74.5	66.0	70.0	72.0	72.2	60	75
9:00	66	73	71.5	71.8	74.5	74.2	65.5	69.5	72.0	72.5	61	
9:10	64	69.4	70.0	71.1	73.8	73.5	64	68.0	70	71.5	59.2	
9:20	63	68	68.5	70	73	73	63.0	67.2	69	70.5	59	
9:30	62.5	68	69.2	70	72.8	72.8	63.0	66.7	69	70.0	59	75

TABLE 5—TEST STARTED WITH ALL HEATERS AND FANS ON. UPPER HEATERS SHUT OFF AT 3:50 AND TURNED ON AT 4:45. JAN. 23, '29, OUTSIDE TEMP. -6° F

TIME	TEMP. ON BALCONIES						C. L. TEMP.			REMARKS
	1	2	3	4	5	6	FLOOR	50 FT. FROM FL.	CEILING	
3:15	67	71.1	72.4	73.	76.1	78.1	59			
3:30	66.8	69.1	71.7	72.5	76.0	77.5	59.5	76	79.1	Top Heaters
4:00	68.2	70	70.8	71.5	72.2	73.0	60.0		73.6	shut off at 3:50.
4:30	67.2	67.5	67.8	68.4	69.5	70.0	61.0		70	Upper Heaters turned on 4:45.
5:00	67.1	68.1	70.7	71.0	74.0	75.5	60.0	77.4		
5:15	67.0	69.1	70.7	72.0	75.0	76.5	60.0			

TABLE 6—ALL HEATERS AND FANS STARTED 1½ HRS. BEFORE TEST. UPPER HEATERS SHUT OFF BEFORE TEST STARTED. FEB. 15, '29, OUTSIDE TEMP. +28.5° F

TIME	TEMP. ON BALCONIES						C. L. TEMP.		
	1	2	3	4	5	6	FLOOR	50 FT. FROM FL.	CEILING
3:00	70	73	73	75.2	76	77	65		80
3:15	71	72	71	74.5	75	75	65		
3:30	71	70.5	70	72.5	73	73	64.5	76	
3:45	71	70.5	69	72.0	72	72	64.5		
4:00	70	70	69	71	71	71	66		71
4:15	70.3	69.3	68.5	70	70.5	70.5	66	70	
4:30	69.3	69	68	70.3	70.5	70.5	66		
4:45	69.8	69	68	69.5	70	69	66		

spaces appear to be from practically uniform from top to bottom to almost any amount warmer at the top than at the bottom depending upon how much heat is supplied at the top in proportion to how fast it is lost through the walls. The temperature gradient may be made practically uniform from top to bottom or it may be otherwise controlled providing any increases are from the bottom upward.

Since in a building of this nature it is desirable to keep the floor level cool and the seats at about 68 deg., one method would be to draw the heating air for the balcony section from some place above the top of the balconies (preferably at the top) and distribute it underneath, letting it filter up through the seats. This would mean that the top of the space would be at balcony temperature or above. If the playing floor became too warm, the lower level could be controlled by letting cooler air in at the floor line. In case the heat losses from the building were not enough to take care of the heat supplied by the audience, it would be necessary to open the exhaust fans at the top and perhaps to take some cold air in the intake sides of the fans. Such a system would only be adapted for cold outside temperatures where heating is required a greater part of the time.

For the system as installed, the better method of control is to regulate the steam supply to the upper heaters to give the required temperature in the upper part of the building to take care of heat losses. The temperature differential betwen the lower and upper balconies may be reduced by the air circulation and by reducing the air discharge temperatures as low as possible and still take care of the heat losses. Temperature regulation can also be gotten by shutting off the upper fans after the building has been brought up to temperature, but this method is not as quick nor does it give as satisfactory a temperature distribution over the balconies as when fans are running. The floor temperatures were disturbed but very little by either method of control.

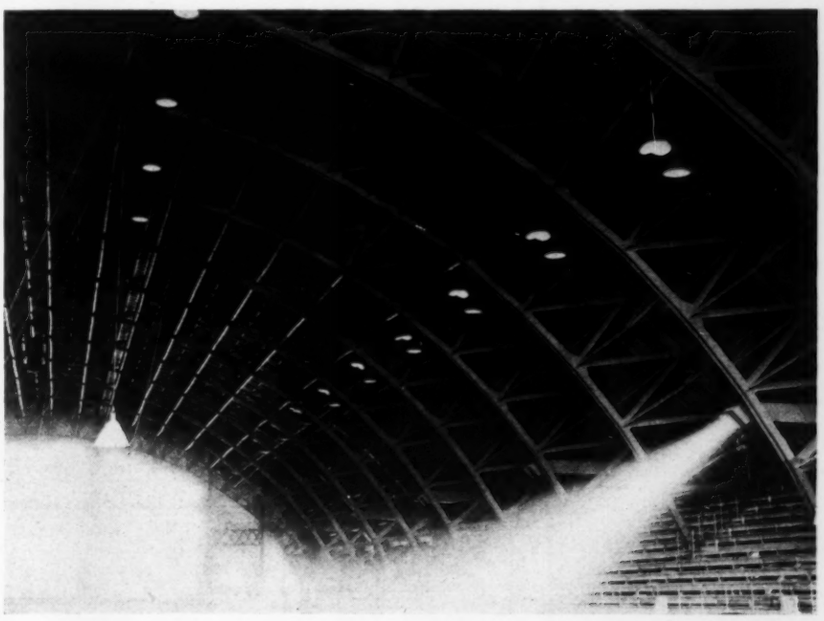


FIG. 6—A SMOKE TEST

The Design of Piping to Secure Flexibility

By Arthur McCutchan*

GROWING realization among industrial power plant designers and operators that very real economies can be effected by the use of higher pressures and temperatures has given a new impetus to general interest in what were once termed theoretical problems in piping.

The development of various state and national codes having to do with the proper installation and safe guarding of piping systems has further emphasized the necessity for a rational analysis of the conditions under which a given pipe line will operate. The requirements of these codes are intended to be constructive rather than restrictive in nature. In their best form they represent a codification of the preferred practice of the best authorities

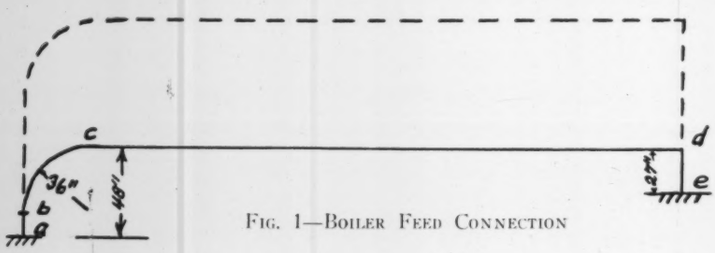


FIG. 1—BOILER FEED CONNECTION

on each phase of a particular type of piping and as such should be carefully studied whether the particular requirements are mandatory or merely recommended practice.

An important section in all piping codes calls to attention the necessity for providing means for taking care of expansion and contraction of the various members of a piping system. For example Section II on Power Piping Systems of the proposed *A. S. A.* Code for the Pressure Piping, contains the following tentative paragraphs.

"THERMAL EXPANSION. Thermal expansion of piping shall be provided for preferably by pipe bends, offsets or changes in direction of the pipe line itself. Pipe expansion bends and steam piping generally shall be cut short approximately 50 per cent of the calculated expansion and drawn into place. Expansion joints of either the slip-sleeve or swivel types may be used, provided however (a) materials used in their construction conform to this Code; (b) the design provides proportions for the structural and working parts ample therefor; and (c) for pressures in excess of 15 lb. the design embodies provision against complete disengagement of the working parts while in service.

"Non-reinforced copper corrugated expansion joints shall not be used for steam pressures exceeding 25 lb. per sq. in. Corrugated copper expansion joints when reinforced with exterior rings or sleeves may be used for steam pressures not exceeding 200 lb. Nothing in this paragraph shall be construed as barring corrugated pipe or bends made of the same material as the remainder of the pipe."

By the very nature of the many variables involved, requirements of this kind are and must remain of a general character. That they are not specific does not mean, however, that they can be disregarded with impunity. The penalties are leaky joints, excessive thrusts on equipment and in some cases difficulty in keeping pipe in position. Dangerously high stresses may exist in certain portions which, while no untoward occurrence may disturb the operation of the plant, do, nevertheless, constitute a possible hazard to life and property.

A Boiler Feed Line

An instance of this kind was encountered in the layout of a boiler feed line leading from a valve nest to a boiler drum as illustrated in Fig. 1. The valve nest was rigidly attached to the building steel so any possible movement of the line at the valve was prevented. The boiler drum, due to its size and method of support, was also relatively immovable. It is rather apparent in this case that each end of the line is so restrained both as to translation and rotation of the ends that they can be considered as fixed. Thus any change in length of the line due to temperature changes can only cause rotations and movements of the elements of the line itself.

The stresses found by a recently published graphoanalytical method ** indicated that the elastic limit of the piping material would have been exceeded while the

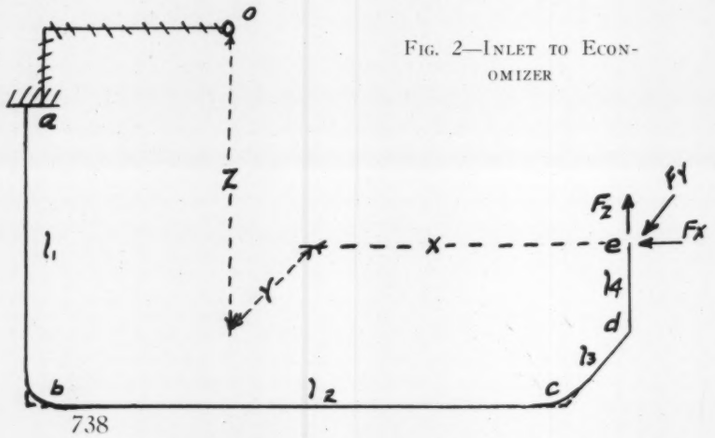
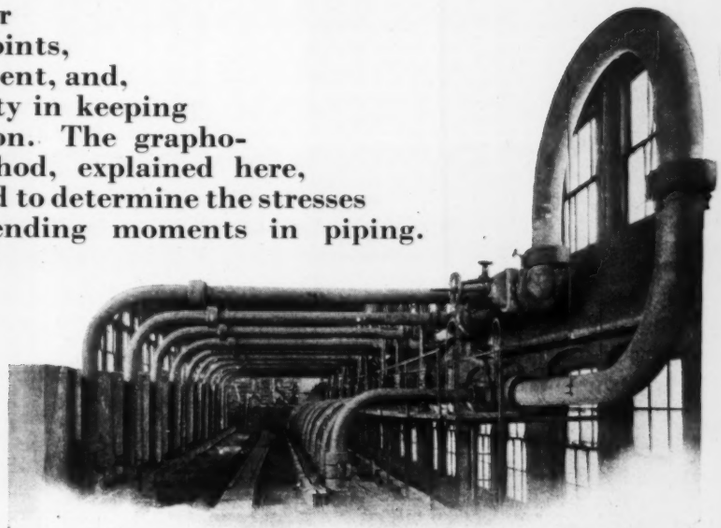


FIG. 2—INLET TO ECONOMIZER

* Engineer, Engineering Division of The Detroit Edison Company.

** Chapter VII, Piping Handbook, Walker and Crocker, McGraw-Hill Book Company.

The penalties for the improper design of piping are leaky joints, excessive thrusts on equipment, and, in some cases, difficulty in keeping the pipe in position. The grapho-analytical method, explained here, may be used to determine the stresses and bending moments in piping.



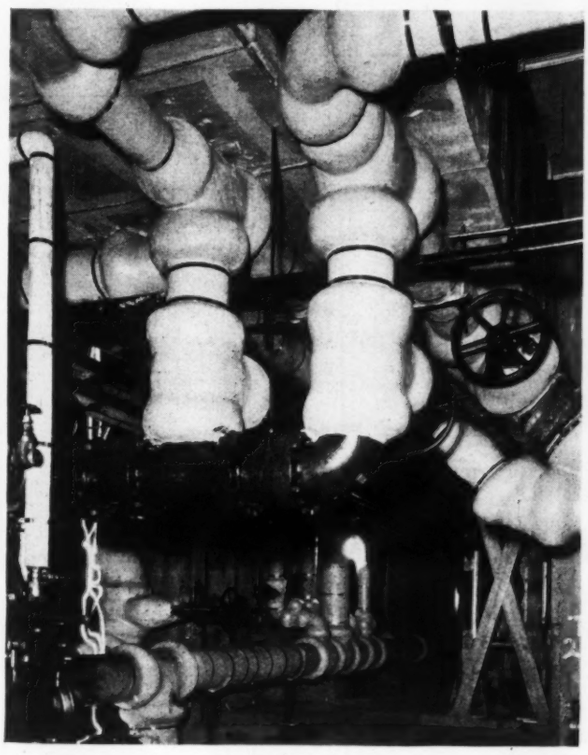
A Glimpse of Some of the Piping in the Boston Terminal Company's Southern Terminal Station

bending moments at the joints would undoubtedly have caused trouble.

The remedy chosen in this case was to feed the boiler through the far end of the drum, thus securing a run of piping at right angles to the original layout. An alternative, had space permitted, would have been to increase the vertical height of the line as shown dotted in Fig. 1.

Inlet Piping to an Economizer

A more difficult problem but one susceptible of ready solution by the method referred to in the foregoing was presented by the inlet piping to an economizer, Fig. 2. In this case the forces and moments acting in three planes were required. The ends of the line were considered as fixed at points *a* and *e*, while the distances over which expansion generated forces and stresses were measured between points *o* and *e*. This layout was found to be satisfactory from a stress standpoint.



High and Low Level Cold Water Supply System from the House Pumps in a Hotel

Grapho-analytical Method Applied to Heating System Supply Line

A different type of problem and one more commonly encountered is illustrated by the heating system supply line of Fig. 3. The line is made of 1½ in. extra strong pipe and carries 200 lb. saturated steam. On first impression it seemed doubtful if the directional changes were sufficient to allow the line to absorb the thermal expansion. As this problem of an unsymmetrical line is one which was formerly only susceptible of solution by complicated methods, quite difficult to apply, it is thought that a detailed analysis of this problem may not be amiss, illustrating as it does the principles of the grapho-analytical method referred to in the previous problems.

The first step is to determine the amount of expansion that will occur when the line is heated from room temperature of about 60 F to the operating temperature corresponding to 200 lb. gage saturated steam of 388 F. The expansion of steel pipe when heated to 388 F is 2.94 inches per 100 feet as determined from the data on ingot steel. (See discussion of coefficients of linear expansion of piping in the October 1929 issue of Heating, Piping and Air Conditioning, p. 529.) The similar figure for 60 F is 0.21 inches. The difference, 2.73 inches per 100 feet, is the amount that must be absorbed by the elements of the pipe line.

The distance over which this expansion occurs is 233 feet in the horizontal direction, designated as X, and 61 feet in the vertical direction, designated as Y, see Fig. 3. The elongation in the X direction is therefore

$$\triangle_X = 2.728 \times \frac{233}{100} = 6.36 \text{ inches, that in the } Y \text{ direction } \triangle_Y = 2.728 \times \frac{61}{100} = 1.66 \text{ inches.}$$

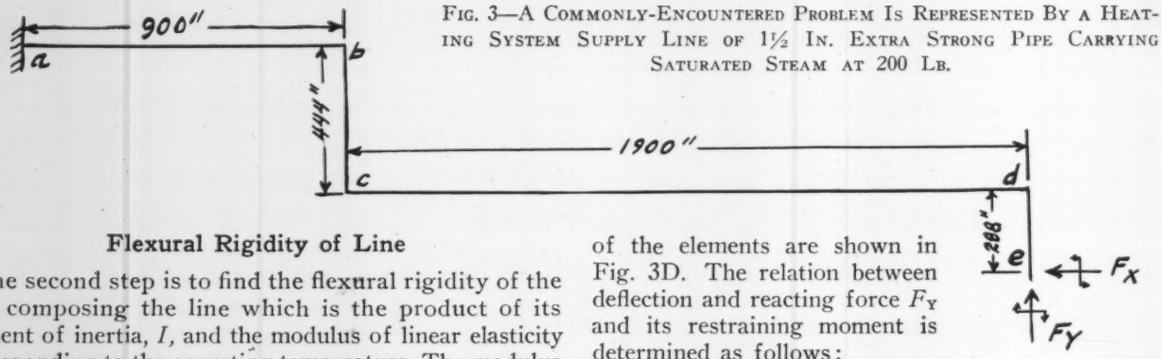


FIG. 3—A COMMONLY-ENCOUNTERED PROBLEM IS REPRESENTED BY A HEATING SYSTEM SUPPLY LINE OF 1½ IN. EXTRA STRONG PIPE CARRYING SATURATED STEAM AT 200 LB.

Flexural Rigidity of Line

The second step is to find the flexural rigidity of the pipe composing the line which is the product of its moment of inertia, I, and the modulus of linear elasticity corresponding to the operating temperature. The modulus of elasticity of steel at 388 F may be taken as 29×10^6 lb. per sq. in. The moment of inertia of a 1½ in. extra strong pipe is 0.3912 inches to the fourth power.

Relations Between Horizontal and Vertical Forces

The third step is to determine the relations which exist between the forces F_X and F_Y and the restraining moments at the ends of the line which result when the attempted elongation of the line is prevented.

A bending moment diagram for the F_X force is drawn as in Fig. 3A. The areas above the base line represent the effect of the F_X force tending to rotate portions of the line in a clockwise direction. Since the line is in equilibrium due to the various forces and moments acting, the magnitude of the restraining moment may be determined by finding the sum of the areas above the line and dividing by the total length of the base line. The algebraic sums of the rotations to the left of each element are found and designated as ψ_b, ψ_c, etc. The sum of the rotations must, of course, equal zero for the entire line as it is in static equilibrium.

The deflection of each element due to the forces and moments acting upon it are illustrated in Fig. 3C for the F_X force and are found as follows:

$EI\Delta_{Yb} = \psi_M 1/2\, l = 283{,}000 \times 450 = 127{,}300{,}000\ F_X$ down
$EI\Delta_{Xc} = \psi_b\, l = 283{,}000 \times 444 = 125{,}800{,}000\ F_X$ left
$\psi_F\, 2/3\, l = 98{,}500 \times 2/3 \times 444 = 29{,}100{,}000\ F_X$ left
$\psi_M\, 1/2\, l = 57{,}400 \times 1/2 \times 444 = 12{,}720{,}000\ F_X$ right
$EI\Delta_{Yd} = \psi_c l = 324{,}100 \times 1900 = 616{,}000{,}000\ F_X$ down
$\psi_M 1/2\, l = 245{,}000 \times 950 = 233{,}000{,}000\ F_X$ up
$EI\Delta_{Xe} = \psi_d\, l = 79{,}100 \times 288 = 22{,}800{,}000\ F_X$ left
$\psi_F 2/3\, l = 41{,}500 \times 2/3 \times 288 = 7{,}950{,}000\ F_X$ left
$\psi_M 1/2\, l = 120{,}600 \times 1/2 \times 288 = 17{,}360{,}000\ F_X$ right

SUMMATION

$EI\Delta_Y$ = 127,300,000 F_X down	$EI\Delta_X$ = 125,800,000 F_X left
616,000,000 F_X down	29,100,000 F_X left
743,300,000 F_X down	22,800,000 F_X left
233,000,000 F_X up	7,950,000 F_X left
$EI\Delta_Y$ = 510,300,000 F_X down	185,650,000 F_X left
	12,720,000 F_X right
	17,360,000 F_X right
	30,080,000 F_X right
	$EI\Delta_X$ = 155,570,000 F_X left

A similar bending moment diagram is drawn for the F_Y force as shown in Fig. 3B. In this case the rotations above the line are counterclockwise but the convention of drawing the bending moment diagram so that the effect of the force acting on the first element is drawn above the base line is used to avoid confusion. The deflection of the elements are shown in Fig. 3D. The relation between deflection and reacting force F_Y and its restraining moment is determined as follows:

$EI\Delta_{Yb} = \psi_F 2/3\, l = 405{,}000 \times 2/3 \times 900 = 243{,}000{,}000\ F_Y$ up
$\psi_M 1/2\, l = 500{,}000 \times 1/2 \times 900 = 225{,}000{,}000\ F_Y$ up
$EI\Delta_{Xc} = \psi_b l = 905{,}000 \times 444 = 402{,}000{,}000\ F_Y$ right
$\psi_M 1/2\, l = 246{,}000 \times 222 = 54{,}600{,}000\ F_Y$ right
$EI\Delta_{Yd} = \psi_c l = 1{,}151{,}000 \times 1900 = 2{,}185{,}000{,}000\ F_Y$ up
$\psi_F 2/3\, l = 1{,}800{,}000 \times 2/3 = 1900 \times 2{,}280{,}000{,}000\ F_Y$ up
$\psi_M 1/2\, l = 2{,}563{,}000 \times 1/2 \times 1900 = 2{,}440{,}000{,}000 F_Y$ down

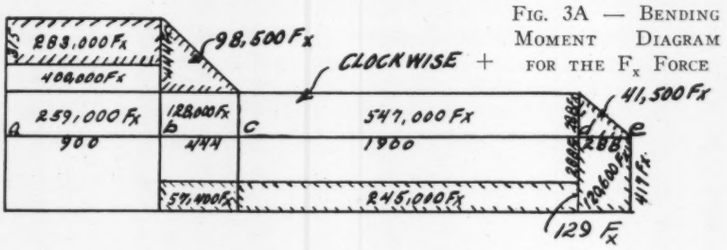


FIG. 3A — BENDING MOMENT DIAGRAM FOR THE F_X FORCE

$EI\Delta_{Xe} = \psi_d l = 388{,}000 \times 288 = 112{,}000{,}000\ F_Y$ right
$\psi_M 1/2\, l = 388{,}000 \times 144 = 56{,}000{,}000\ F_Y$ left

SUMMATION

$EI\Delta_Y$ = 243,000,000 F_Y up	$EI\Delta_X$ = 402,000,000 F_Y right
225,000,000 F_Y up	54,600,000 F_Y right
2,185,000,000 F_Y up	112,000,000 F_Y right
2,280,000,000 F_Y up	568,600,000 F_Y right
4,933,000,000 F_Y up	56,000,000 F_Y left
2,440,000,000 F_Y down	$EI\Delta_X$ = 512,600,000 F_Y right
$EI\Delta_Y$ = 2,493,000,000 F_Y up	

The deflections to the right and left and up and down of the various elements are summarized and equated to the values of flexural rigidity determined in the foregoing.

(1) $EI\ \Delta_X = 155{,}570{,}000\ F_X$ (left) $\quad 512{,}600{,}000\ F_Y$ (right)
(2) $EI\ \Delta_Y = 510{,}300{,}000\ F_X$ (down) $\quad 2{,}493{,}000{,}000\ F_Y$ (up)

By letting all deflections in the direction of the forces be designated as positive and those in opposite direction of the forces as negative, equations (1) and (2) can now be written as follows:

(1) $EI\ \Delta_X = 155.6 \times 10^6\ F_X - 512.6 \times 10^6\ F_Y$
(2) $EI\ \Delta_Y = -510.3 \times 10^6\ F_X + 2{,}493.0 \times 10^6\ F_Y$
$E = 29 \times 10^6 \quad I = 0.3912^{\text{in}^4} \quad \Delta_X = 6.36'' \quad \Delta_Y = 1.66''$
$EI\ \Delta_X = 29 \times 10^6 \times .3912 \times 6.36 = 72.2 \times 10^6$
$EI\ \Delta_Y = 29 \times 10^6 \times .3912 \times 1.66 = 18.85 \times 10^6$

Substituting values of $EI\triangle_X$ and $EI\triangle_Y$ in Equations (1) and (2) respectively we now obtain the following simultaneous equations, which upon solution will determine the numerical values of the reacting forces F_X and F_Y.

(1) $72.2 \times 10^6 = 155.6 \times 10^6 F_X - 512.6 \times 10^6 F_Y$
(2) $18.85 \times 10^6 = -510.3 \times 10^6\ F_X + 2{,}493.0 \times 10^6 F_Y$
$255.8 \times 10^6 = 823 \times 10^6\ F_Y$
$F_Y = 0.31$ lb.
$F_X = 1.48$ lb.

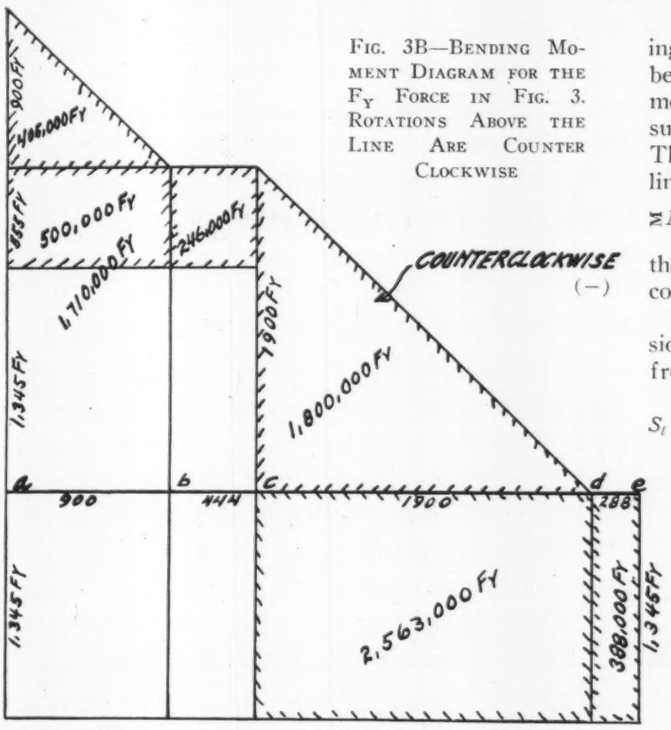


FIG. 3B—BENDING MOMENT DIAGRAM FOR THE F_Y FORCE IN FIG. 3. ROTATIONS ABOVE THE LINE ARE COUNTER CLOCKWISE

The insignificant value of the reacting forces indicates that an analysis of this particular line was scarcely warranted but this is due to the very small moment of inertia of a 1½ in. line. If this particular line had been made of 12 inch extra strong pipe the forces would have been increased in the ratio of the moments of inertia as follows:

$$F_{Y12}'' = \frac{361.5}{0.3912} \times 0.31 = 287 \text{ lb.}$$

$$F_{X12}'' = \frac{361.5}{0.3912} \times 1.48 = 1{,}365 \text{ lb.}$$

The maximum stress in the line is found by substituting the values of the reacting forces directly into the bending moment diagrams. The maximum bending moment in the line, which exists at point c, is found by summing up the F_X and F_Y moments for both diagrams. The following computations are required for the 12 inch line at point c:—

$$\Sigma M_c = (1345 - 1900)\, F_Y + (288 - 417)\, F_X$$
$$= -(555 \times 287) - (129 \times 1365) = -335{,}270 \text{ inch lb.}$$

the minus sign denoting that the bending moment acts counter-clockwise.

The maximum stress due to bending exists at the outside of the pipe wall at a distance $r_o = 6.375$ inches from the neutral axis and is computed as follows:

$$S_t = \frac{M_c r_o}{I} = \frac{335{,}270 \times 6.375}{361.5} = 5{,}900 \text{ lb. per sq. in.}$$

The longitudinal stress due to internal pressure which is directly additive is

$$S_2 = \frac{PD}{4t} = \frac{200 \times 12.75}{4 \times 0.50} = 1{,}270$$

The maximum total stress in the 12 inch line is therefore 7,170 lb. per sq. in., which is within allowable stress limits for this temperature. The bending stress in the 1½ in. line is only

$$S_t = \frac{362 \times 0.95}{0.3912} = 880 \text{ lb. per sq. in.}$$

and that due to internal pressure

$$S_2 = \frac{200 \times 1.90}{4 \times .20} = 475$$

The total stress is therefore only 1,355 lb. per sq. in.

The reason for questioning the flexibility of the layout originally arose from experience with larger lines

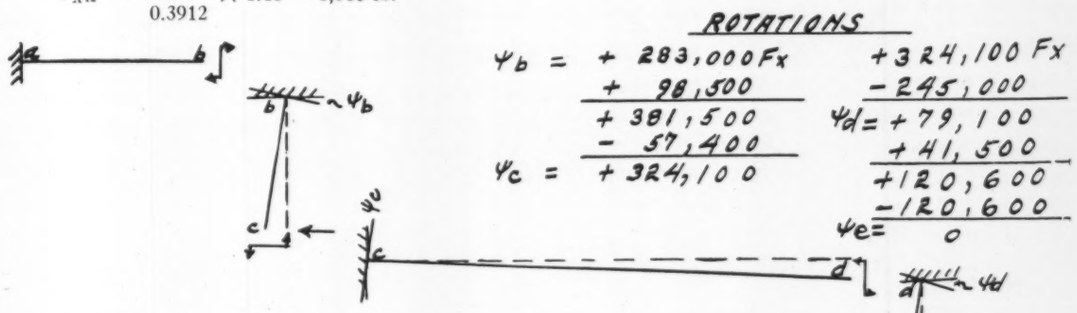


FIG. 3C (ABOVE)—THE DEFLECTION OF EACH ELEMENT DUE TO THE FORCES AND MOMENTS ACTING ON IT FOR THE F_x FORCE

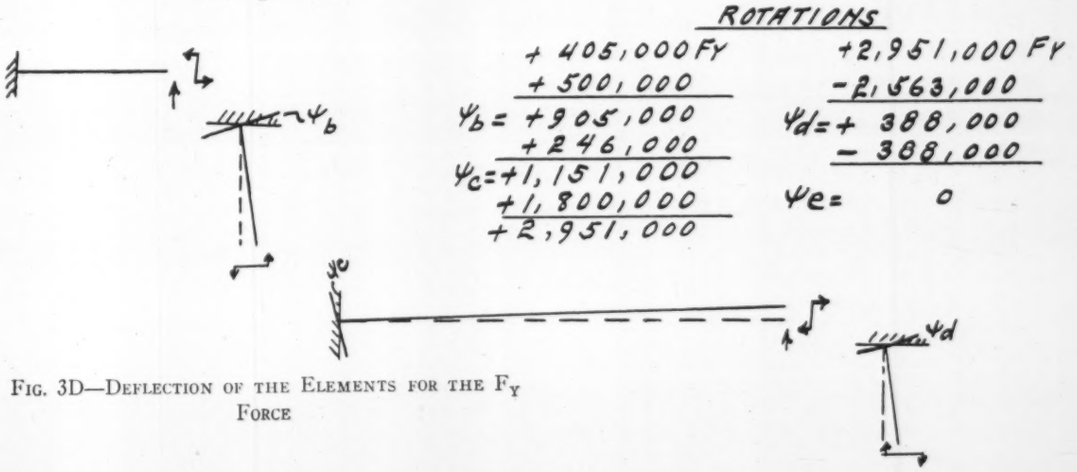
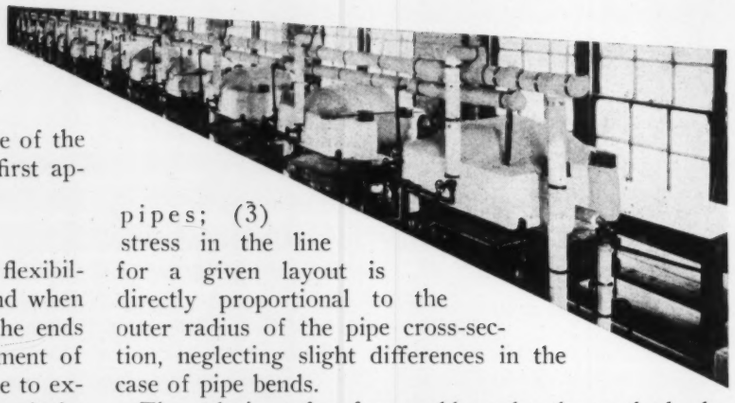


FIG. 3D—DEFLECTION OF THE ELEMENTS FOR THE F_Y FORCE



13 of 26 Press Heads in the Mechanical Rubber Co., Chicago

operating at higher temperatures. The magnitude of the greater flexibility of the small line was not at first appreciated.

General Rules

Certain generalizations may be made regarding flexibility of a piping layout and should be kept in mind when considering a proposed line: (1) reactions at the ends of the line are directly proportional to the moment of inertia of the elements of the line; (2) stress due to expansion in a given size of line is independent of the wall thickness as the moment of inertia cancels out between the expressions for stress and deflection. This is strictly true only for piping composed of straight lengths of pipe as certain modifying factors apply to curved pipes; (3) stress in the line for a given layout is directly proportional to the outer radius of the pipe cross-section, neglecting slight differences in the case of pipe bends.

The solution of a few problems by the methods described above will give a better understanding of the factors which determine adequate flexibility in a piping layout and will represent time well spent for one engaged in the design of piping for high temperature service.

"It's Not the Heat But the Humidity"

AIR at any given instant has two different but equally true temperatures. One of them is that indicated by the dry bulb thermometer and the other is that indicated by the wet bulb thermometer. Human comfort more nearly follows the indications of the wet bulb thermometer than it follows the indications of the dry bulb thermometer. The thermometer, of course, indicates the *intensity* of the heat but not the amount.

Heat is a positive condition. It escapes to some extent from the best enclosures no matter how hard we try to hold it. In any room, if we keep the thermometer warm enough so that it indicates an even or stable temperature, we may be sure we are constantly pumping in heat to replace the leakage (assuming colder air outside).

The amount of heat is expressed in thermal units or, since the British government is commonly supposed to have first established the unit, we use the term British thermal units, or Btu. A Btu is the amount of heat necessary to raise one pound of water one degree F.

The capacity of air to absorb heat is a function of the amount of moisture present in the air at the time. For instance, one pound of air may require anywhere from 24 Btu to 68.8 Btu, without changing the temperature of the pound of air as indicated by a dry bulb thermometer.

If we attach a piece of silk to the bulb of the thermometer and moisten the silk we have a wet bulb thermometer. If the air surrounding this wet bulb thermometer is moved over the moist surface, part of the moisture in the silk will be evaporated into the air. The rate of this evaporation is not a function of the movement of the air but of the relative dryness of the air. If the air moving over the bulb is very dry, the moisture evaporates quickly. When moisture evaporates, heat enters into the process and this heat, doing the work of changing water into vapor no longer exists as heat, so that the wetted bulb of the thermometer becomes cooler than the dry thermometer bulb.

Interestingly enough, the amount that the wet bulb is cooler than the dry bulb alongside it at the same instant is an accurate index of how much moisture there is in the air.

Relative humidity, therefore, may be determined by swinging a wet thermometer bulb alongside a dry thermometer bulb. The instrument based on this fact is called a psychrometer. The greater the depression of the wet bulb below the dry bulb the lower will be the relative humidity of the air.

The human body, like the bodies of all mammals, is much like a wet bulb thermometer. It is constantly regulating its own temperature, principally by sending more or less moisture to be evaporated from the skin, so as to accelerate or to slow down the rate of heat loss. This balances the heat input which, with varying rates of labor or varying rates of metabolism (which means rate of living), may have wide fluctuations but which cannot be allowed to cause any particular change in the body temperature.

If the body temperature of a warm-blooded animal increases, he has fever and is ill, while, if it decreases, he has a chill and is equally ill.

So when we put this wet bulb body of ours in a warm dry room, the evaporation from the skin is rapid, and the room must feel cooler than it really is as indicated by a dry bulb thermometer; and if we put the body in a warm damp room, the evaporation from the skin is slow and the room must feel even warmer than it really is, and we say, "It's the humidity, not the heat," for the dry bulb thermometer doesn't tell the whole truth.

By regulating the moisture content so that it is low in a room with 80 degrees dry bulb and high in another room with 60 degrees dry bulb, people may pass from one to the other without realizing any difference whatever in the temperature.

We have no definite knowledge that one of these conditions, within reasonable limits, is any worse than the other so long as the persons who endure them are comfortable and at ease.

It is believed by many physicians that the lower temperatures and higher relative humidities are in the long run better, since the high rates of evaporation of moisture from the mucous membrane of throat and nostrils in the dry room may cause some irritation and a tendency to a lower resistance to infection. This has not been proved definitely.

Locomotive Smoke Removal Through High Building

REMOVAL of smoke from railroad terminals has been a problem for many years. It is solved in some instances by the use of electric locomotives and in other instances by discharging the smoke into high vaulted sheds or through louvers in low roofed sheds.

We are coming now to encounter a similar problem in office and in loft buildings. Under the Daily News building in Chicago, for instance, are twelve tracks of the Pennsylvania, the Chicago, Milwaukee and St. Paul and other railroads which run into the Union Station from the north. Puffing locomotives are passing through this tunnel every hour of the day. A similar situation may be found in the Merchandise Mart, Chicago, the largest building in the world.

There are few installations such as the one in the Daily News building. Consequently there were no test data on which to base the size of the fans, smoke chambers, stacks and other equipment for this job. It has functioned satisfactorily, however, for a number of months. This smoke removal question seems to be a western problem, since air rights buildings in the east have usually been over stations into which electric locomotives are run.

Fig. 1 shows the track space, smoke chamber and stack. Slots in the ceiling of the track space take the smoke directly from the locomotives into the smoke chamber *AA*, the typical section of which is shown. The slots in the chamber are continuous except for some structural members crossing the entire track area. The structural members, and the driveway which extends the entire length of the building from north to south, form an obstruction that restricts the passage of smoke to some extent from the northwest corner of the track area to the stack. The smoke chamber has several high portions from which connections are made to the stack. The large space provided by the smoke chambers permits the expansion of the gases and prevents the downward surge which otherwise would follow each blast into the chamber from a locomotive.

The stack has a cross-sectional area of 380 sq. ft. and is divided by a partition which forms two stacks, one side for gravity and the other for fan exhaust. The gravity division has a cross sectional area of 220 sq. ft. and the fan division a cross sectional area of 160 sq. ft. A 15 hp. fan was installed to draw the smoke from the isolated northwest corner area and discharge to the stack and it was intended that the remainder of the area should be ventilated by gravity. It has been found, however, that it is not necessary to use the fan except under unusual conditions. The entire stack area is used as a gravity stack except when unusual weather or unusual amounts of smoke make it necessary to use the fan. At such times, periodical running of the fan for short intervals has proved sufficient.

The system was designed to discharge approximately 195,000 cubic feet of air per minute as a maximum requirement. Of this 35,000 cubic feet per minute is fan capacity and 160,000 cubic feet per minute was assumed as the capacity of the gravity system.

All structural steel members in the smoke chamber are protected with concrete. The baffles are of cast iron. The fan is housed in masonry and discharges through masonry. The fan itself is protected from the gases by asphalt paint.

As shown in the drawing, provision is made for a future exhaust fan at the top of the gravity stack if more positive exhaust from the smoke chamber should become necessary.

Holabird and Root of Chicago were the architects for the building and J. A. Sutherland was the mechanical engineer for the architects.—*Thomas R. Shaver.*

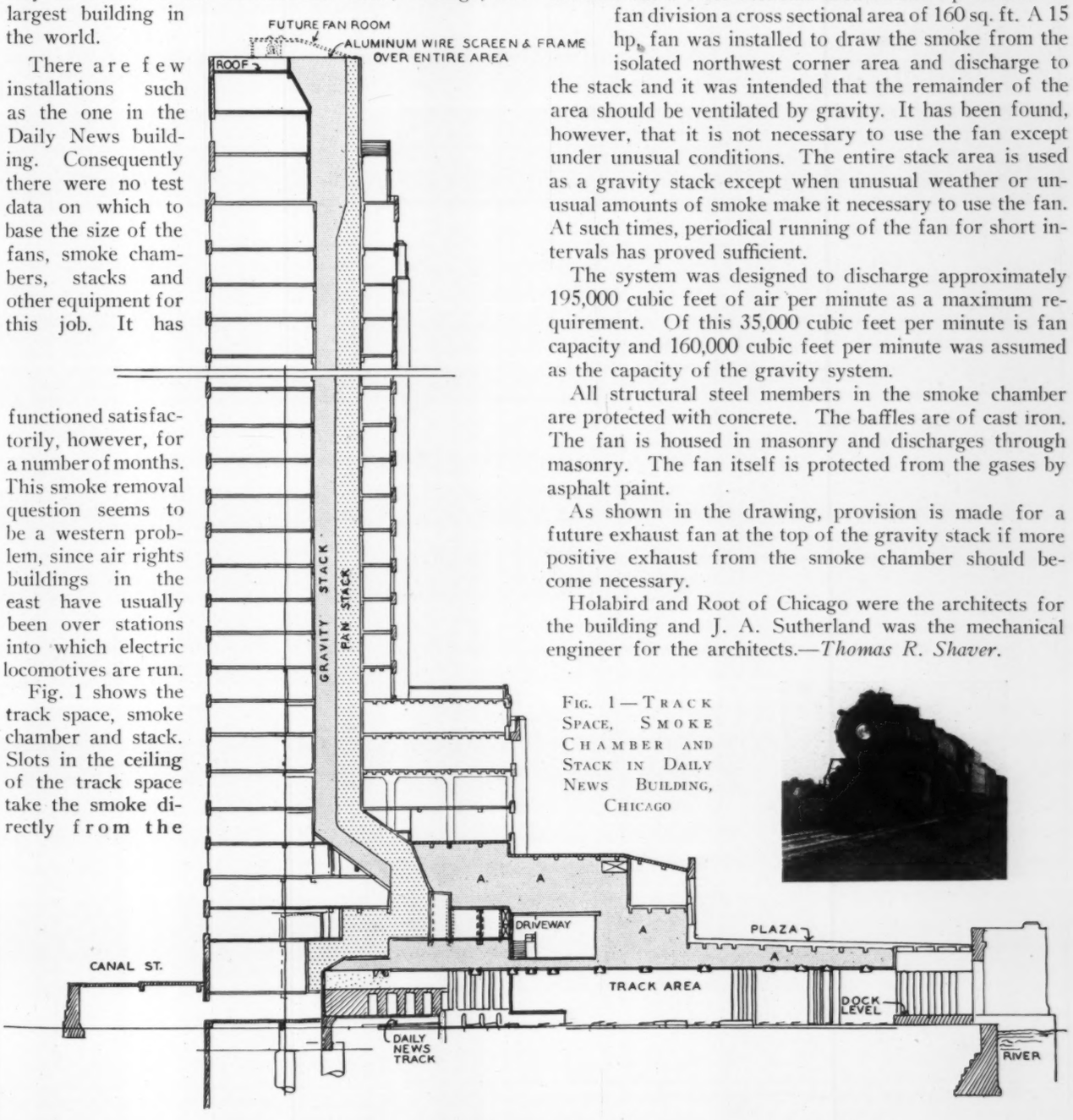


FIG. 1—TRACK SPACE, SMOKE CHAMBER AND STACK IN DAILY NEWS BUILDING, CHICAGO

Suggested Reforms in Heating and Air Conditioning Nomenclature

By Samuel R. Lewis*

Editor's Note: A letter from the author of this article has been received, as follows:

***"The American Society of Heating and Ventilating Engineers* has under consideration the appointment of a committee to develop improvements in nomenclature and its report eventually may be submitted to the Society for adoption."**

"I am presenting this unofficial list of abominations and alleged abatements to manufacturers and engineers and to the public in the hope that my friends will criticise and improve and suggest very freely. If their reactions are interesting they may be printed. In any event, all helpful criticisms will reach this committee of the Society whenever it shall be appointed. Please write to HEATING, PIPING AND AIR CONDITIONING.—Samuel R. Lewis."

FOR a long time I have wished to take a shot at the nomenclature used in the heating and air conditioning fields.

No small part of the pleasure in doing this will be in the controversies which may develop, and in the response which it may evoke from friends who are not in agreement. I do not apprehend that any of my friends will disagree with the proposition that reform in nomenclature is desirable, but I am quite sure no engineer will accept, without a struggle, a new name proposed by anyone but himself for his own pet product.

I believe that if an improvement in the nomenclature used in heating and air conditioning ever shall be achieved it will be as the result of the labor of some exceedingly well-endowed committee, after many meetings and discussions.

In the meantime, however, the subject ought to be ventilated, and so here am I with my shotgun.

There is no intention to enter into profound basic engineering changes which naturally would involve action by technical societies.

Incidentally, however, I wish that we could adopt the metric system, and wish to express my compliments to the manufacturer of heating equipment who has for some years printed the metric equivalents in parallel columns to the standard dimensions in his catalog. He is helping toward adoption of a logical improvement by a process of gradual infiltration.

Perhaps improved nomenclature must come in the same manner.

Aside from this metric system comment the serious propositions herein will be limited to common trade words which might be changed or defined by unofficial custom or general usage.

Taking a current trade magazine let us try to knock over the outstanding offensive words as they seem to protrude from the pages in both advertising and editorial sections.

1. *Horsepower:*

A fan manufacturer's advertisement uses this. Why not let old Dobbin rest in peace where he is buried under the orchard trees? The *power* required by the fan is all it is necessary to state, and power means watts or pounds of steam, and we should think and speak in these terms.

I suggest that we become watt-minded.

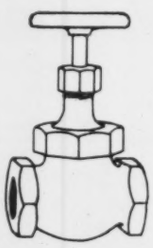
It will be far easier to transform clumsy and ill-defined horsepower to watts or pounds of steam in *every* case.

Several fan manufacturers now are publishing the power requirements of their fans in watts only. Blessings on them!

2. Assembly of fan and heater of small size for heating by recirculation within the room, called popularly a "unit heater":

The term "unit heater," however, is often confused with "heating unit," which may mean any unit of heat transmitting surface. Some name which is descriptive and which permits of easy modification to differentiate between the various heat sources, as electricity, steam and hot water, is needed.

* Consulting engineer, Chicago.



Size	Inches	⅜	½	¾	1	1¼	1½	2
Size	Millimeters	10	13	19	25	32	38	50
Price	Each	1.50	1.85	2.20	2.80	4.10	5.50	8.50

FIG. 1—METRIC EQUIVALENTS OF THE ENGLISH SYSTEM ARE VALUABLE IN CATALOGS OF EQUIPMENT

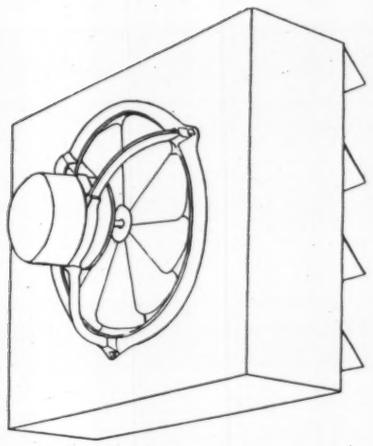


FIG. 2—A STEAM ELECON

The English language apparently has no terse idiom which will serve. To express what one has in mind in this case requires many words and a diagram.

One manufacturer has coined a proprietary word, and nowhere in his advertising can you find him calling his fan-equipped device a "unit heater."

One engineer calls this particular assembly a "unit heating machine." This term seems clumsy and does not indicate the source of heat.

I suggest:

First word indicating the heat source—as steam.

Second word—

First syllable—indicating the fan power—ele—.

Second syllable—indicating the type of heater—con.

Thus a *steam elecon* is a unit heating machine composed of the assembly of a steam-supplied, convector type heater and an electric-driven recirculating fan.

Elecon in general is a new noun meaning such a device. If the fan is driven by a steam turbine it is a *steam steamcon*. An *electric elecon* or a *gas elecon* tells its own tale.

3. *Assembly of fan and heater of small size for ventilating or recirculating:*

This is the type, as distinguished from the one just before discussed, which has some sort of valves or dampers for varying and controlling the temperature of the delivered air and which usually takes air from outside. It is sometimes called a "unit ventilator," which also may mean a single small cowl on a roof for letting air pass and keeping out rain. One engineer calls it a "unit ventilating machine."

I suggest:

First word, indicating the heat source, as steam.

Second word—

First syllable, indicating the fan power, ele—.

Second syllable, indicating the function, vent.

Thus an *elevent* is an electric driven unit ventilating machine, and a *hot water elevent* is an obvious device from its name.

4. *Encased heaters performing the greater part of their service by transfer to the passing air of the heat, rather than by radiation:*

They do not depend on fans for air movement, as do *elecons*, but depend on the current induced by the heated flue above them. Their capacity is partly proportional to this draft head. They are called "concealed radiators," "encased heaters," and the *Guide* of the *American Society of Heating and Ventilating Engineers* calls them "cabinet convectors."

No one knows just what is meant when any of these terms is used without a diagram, or until, happily, the name of some well-advertised manufacturer of this type of heater is mentioned. The device surely is a convector. What we need is some inspired name for it which describes it; then a campaign of education until everyone uses the new term.

To be consistent with *elecon*, the second syllable may be "con."

I suggest—*fluecon*.

A recessed *fluecon* is built into the wall, while an exposed *fluecon* stands free.

5. *Fan system heat surface:*

This ponderous term is required to convey the information.

The *American Society of Heating and Ventilating Engineers' Guide* suggests "blast indirect convector," which is just about as happy as putting sugar in one's tea with a scoop shovel.

"Blast radiation," the older term, is poor, because it isn't radiation, and the term in any event, is inelegant. We used to call these heaters coils, long after we ceased to use steel pipe-risers screwed into cast iron bases, which were never fairly to be dubbed "coils" anyway.

The stuff certainly is used as a convector, not as a radiator.

I nominate—*fancon*.

Fancons may be used with induced air currents no matter what power is used to drive the fan. A *fancon* may be used to warm the air at the base of a heated flue without any fan at all. A *fancon* is the heating element used in an *elecon* or in a *fluecon*.

6. *A fluecon depends on the height of its flue for its capacity. Until everyone learns this thoroughly there will be occasional troubles with the fluecon installations:*

"Draft-head" seems to be an acceptable term for this flue-height. The heating elements of various makes of *fluecons* vary in height. The higher these heating elements are, generally, the less work there is done by the upper parts of them, since the air has already been warmed before it can touch them.

In order that uniformity of practice and assurance

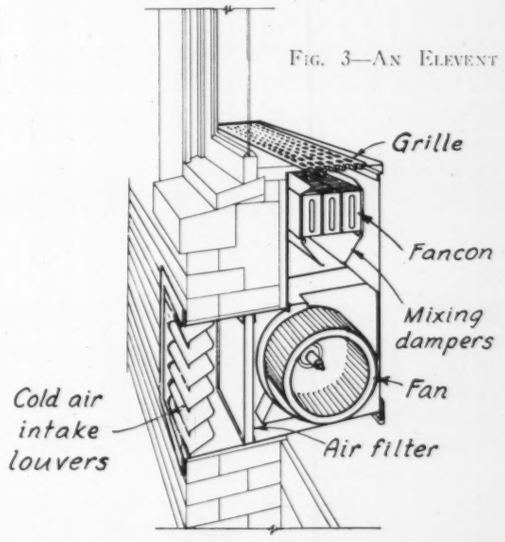


FIG. 3—AN ELEVENT

of results shall be attained in this comparatively new type of heating apparatus, I suggest:

Draft-head means the vertical distance from the *top* (not the bottom) of the heating element in a *fluecon* to the bottom of the outlet opening from the *fluecon.* If the *fluecon* looks up rather than out, the draft-head is the distance from the top of the heater to the top of the *fluecon.* (Note this dimension marked in the drawing of a *fluecon.*)

7. *Thermostatic apparatus placed at the return end of a steam-using heater to let water and air pass out to the return pipe and to interfere with the passage of steam:*

Some manufacturers call this a "valve;" others call it a "trap." Some call it a "vacuum valve" or "vacuum trap."

In order to differentiate between the types, the term "thermostatic trap" is used, as opposed to float trap. The term "float trap" covers the larger types of trap having floats and levers, and leads to confusion, while when the term "thermostatic valve" is used, one never knows without a diagram whether the speaker means the outlet device or a supply valve for the heater, controlled by a thermostat.

I suggest: *Thermotrap.*

This term is not proprietary, as far as I can learn, and says reasonably well just what it means, as differentiated from a float trap.

Since the same trap may be used for a gravity-return-steam system as for a pump-return-steam system, we need not bother with vacuum trap or vapor trap, but can call it a *thermotrap* and can thereby understand what we mean.

8. *We come to systems of heating. What do we think we mean when we say "vacuum system," "vapor system," "vacuum-vapor system," etc?*

I know that I now tread on ground quaking with subterranean fires and liable to break forth into heavy explosions.

The state of the nomenclature however is indeed sad, and one cannot forbear making an attempt to improve it.

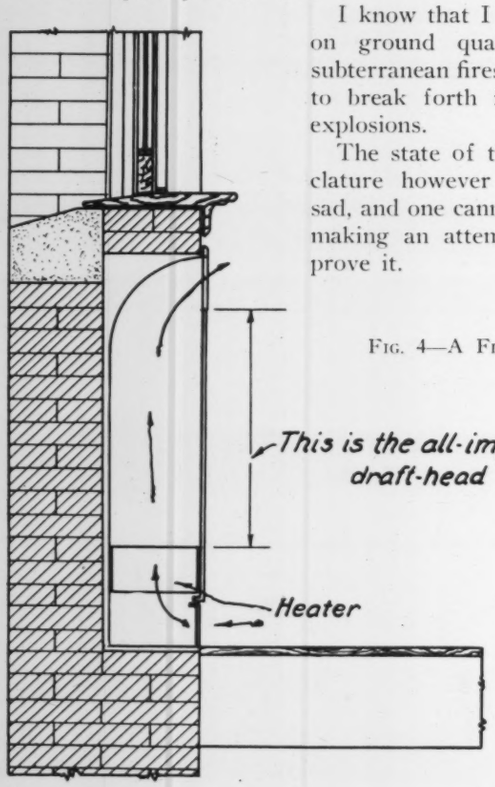


Fig. 4—A Fluecon

The *Guide* does not give much help. The *Guide* needs a clean broom in this particular. It seems to me that we might agree that vacuum and vapor systems are essentially the same, using the same general piping, valves and traps; the vacuum system having a power driven pump while the vapor system has gravity returns.

If the plant has no power driven pump it must use gravity returns, even though these may involve tilt traps or alternating receivers or equivalent devices.

Any gravity return system of steam heating may develop a very respectable sub-atmospheric pressure.

I suggest:

(a) *Direct-vent systems.*

This may be single-pipe or double-pipe, and may have gravity return or pump return. It is essentially a system of steam circulation which works in cycles, generating enough pressure to eject the air, then letting the pressure run down preparatory to another cycle. It means, what the *Guide* apparently means by its confusing term, "gravity system." The use of non-return air valves does not change such a system from being a *direct-vent system,* though it could be called a *direct vent vacuum system* and be truthful.

(b) *Vacuum system.*

This is a two-pipe system with the heating elements equipped with outlet-traps which drain to a power-driven pump, or its equivalent, which removes the air and the condensation.

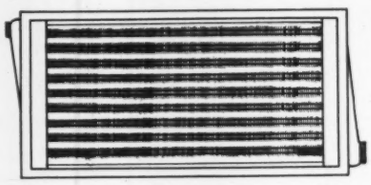


Fig. 5—A Fancon

(c) *Vapor system.*

This is the same as (b) except that some air eliminating device other than a pump removes the air while the condensation returns by its own weight to the boiler, or to a receiver, which must be lower than the radiation. If the plant uses a condensation pump rather than a vacuum pump it is still a vapor system rather than a vacuum system.

9. *Insulation* is a term much to be preferred to "pipe covering."

10. Suppose we agree to let *valves* refer to devices which control materials which are carried in pipes and tubes such as steam, water, compressed air, and that we agree that devices in air ducts and flues shall be called *dampers* rather than *valves.*

11. A *register* is the combination of perforated face plate and damper. The thing into which the dampers fit and which is clamped to the face plate is the *register body.* A *grille* is the face plate without any damper.

12. When we insert a thermometer into any duct or pipe we should have a receiver for it, so that the thermometer can be removed easily for testing or replacement. This inserted cup or socket or well is often difficult to describe without a diagram.

I suggest: *Bulb chamber.*

We can so easily thereafter call for these where they are needed without danger of being misunderstood.

13. Is a *filter* properly to be called after the material which it removes, or after the material which remains following the action of the filter?

All breathers have a very useful instrument in their mucus-coated nose hairs which filter out the dust. The ventilating industry has adapted this idea to filters which clean the air used for ventilating purposes. They are called *dust filters, air filters, oil filters,* etc., until one seldom knows what is meant.

When we speak of water filters we create a precedent. We do not speak of mud filters, so I deduce that it should be proper to call the filter after what remains following the action of the filter.

I nominate the term *air filter,* therefore, and hope that all of the manufacturers will come along. An oil filter is used for filtering oil and for no other purpose. Nobody knows what good is a dust filter!

14. When we use a *steam-supplied hot water heater* we may employ either of two general types; one in which the water is around the steam tubes; the other in which the steam is around the water tubes. The physical construction of the interchanger may be the same or very little different, for either method. To tell what is wanted in this case is always rather awkward. Some manufacturers differentiate by arbitrary symbols, as type *K* for one and type *R* for the other.

I suggest the adoption in this case of *steam tube* and *water tube* as the designating terms.

For heating the pumped, filtered, chlorinated, recirculated water of a swimming pool, therefore, I'd specify a *water tube* heater.

15. One should not get away from this vein without inverse compliments to *square foot.*

I suggest that every manufacturer of heating apparatus forthwith begin to rate his boiler or his *elecon,* his *elevent,* his *thermotrap,* his *fluecon* or his *fancon* in heat units, and that at least he mention these in parallel with the obsolete square feet.

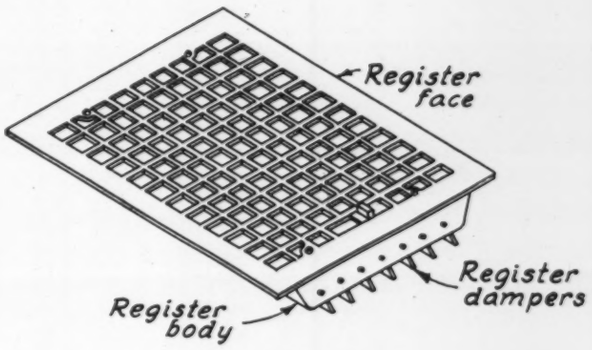


FIG. 6—A REGISTER

The argument against this is made that heating is a very slowly-progressing art, and that failure to name square feet will alienate the affections of some old time fitter like Sam Lewis, who can think only in terms of old time square feet.

The old time fitter will be less likely to make a mistake, I submit, if the poor doddering old soul with his stub pencil and scrap of paper or with his piece of chalk on the side of a boiler doesn't have to transpose and translate his "square feet" too many times on the way from building-heat-loss to boiler-outlet. In the whole process of carrying the heat from the coal through the fire to the steam, through the pipes to the heaters and out through the cooling surfaces of the house, the only square feet which have any business in the picture are those of the glass and wall and other heat-losing areas. All the other very simple computations can be made much less arduous by keeping to heat units and heat units alone.

Picture the absurdity of figuring the heat loss from the building in the only intelligent and intelligible way there is, via heat units, then dividing some of it by 240 to get square feet of radiation, then multiplying

GLOSSARY OF NEW TERMS SUGGESTED

(1) Power—Horsepower. Let us do away with the horse.
(2) Elecon—Recirculating fan-equipped unit heating machine.
(3) Elevent—Unit ventilating machine which takes its supply of air from out-of-doors.
(4) Fluecon — Gravity cabinet convection heater; any heater placed in a flue and recirculating by gravity.
(5) Fancon—Blast indirect heater.
(6) Draft-head—Vertical distance from the top of the fluecon to the bottom of the outlet opening for the air which passes around the fluecon.
(7) Thermotrap — Thermostatic trap used with vacuum systems or with vapor systems.
(8) Direct-Vent System—Any steam heating system which uses air valves which discharge directly from the heater to the atmosphere. Let us call the single-pipe system and its name both obsolete.
(9) Vacuum System — Any steam heating system which uses thermotraps and a pump.
(10) Vapor System—Any steam heating system which uses thermotraps and gravity return of the condensation to the boiler or to the receiving tank.
(11) Insulation—To be used instead of "covering."
(12) Valves—Used on pipes but not in ducts.
(13) Dampers—Used on ducts.
(14) Register—Combination of register face and register body.
(15) Register Body — Dampers and body which usually are attached to the face.
(16) Bulb Chamber—Thermometer receptacle for ducts or pipes.
(17) Air Filter—Air cleaning device, as differentiated from a dust filter.
(18) Square Foot—An obsolete term.

some of this by 240 to get heat units for selecting *elecons* or unit heaters, and dividing some of it by 966, perhaps, to get pounds of steam for selecting *fancons* or heaters for use with basement-located supply fans, and finally, having picked out the various heat transfer agents again multiplying the square foot rating of each by its heat unit output, then dividing this sum by 240, then adding a mysterious safety percentage in order to select the boiler.

Opposed to this foregoing terrific sentence we would measure the square feet of each kind of heat losing surface, and multiply each by its heat loss factor. The total heat loss in heat units tells directly the boiler size when this is rated properly in heat units. The heat loss in each room, say 11,580 heat units, gives directly the output required for the heater in each room and the plans would be marked: "Number So and So heater, 12,000 heat units."

The sizes of the piping, traps, valves, etc., likewise would be selected easily and quickly from tables based on heat units transmitted, rather than on square feet served.

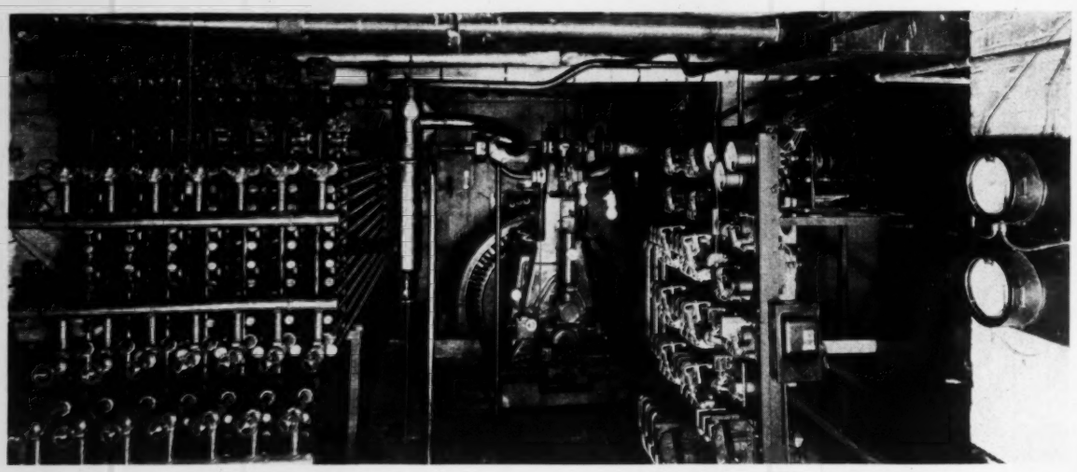


FIG. 1—COMPRESSOR, CONDENSER AND SWITCHBOARD IN THE HOTEL SINTON, CINCINNATI

Modern Air Conditioning Systems Increase Patronage of Older Hotels

By H. L. Branigin*

STATISTICS on building operation show rapid reduction in rental values in all types of buildings. As methods of building construction are improved and as mechanical equipment becomes more efficient, the older buildings lacking the newer equipment bring in less rental returns. This deterioration is especially rapid in hotel buildings. However, many buildings that are fine examples of architecture and construction may be brought up to date and made strictly modern by improvements in the heating, air conditioning and other mechanical equipment at a reasonable expenditure.

Hotels and dining rooms frequently build up a distinct commercial asset with age, since most people prefer to patronize establishments where the quality of food and service is well known to them, and the rooms themselves may recall pleasant memories and associations. On the other hand, such patronage demands modern equipment and the best of sanitation and air conditioning.

The Hotel Sinton, in Cincinnati, Ohio, offers a notable example of this. Modern equipment helps to retain old patrons and to attract new ones. This hotel installed an air conditioning apparatus for the Grand cafe and a large coffee shop. The results obtained have adequately demonstrated the wisdom of the investment.

Fosdick and Hilmer, mechanical engineers, in laying out the new equipment for the Hotel Sinton, followed approved practice in the selection and installation of apparatus in order to secure the results desired with most satisfactory and economical operation.

Separate systems were provided for the cafe and the coffee shop served by a single refrigerating unit. The CO_2 compressor is a two-cylinder vertical, double-acting special air conditioning type, having a capacity of 120 tons and driven by a 175 hp. synchronous motor running at 225 rpm. The condenser is a two-pipe safety welded type, 20 ft. long, 14 sections, 10 pipes high. The installation includes a switchboard with all necessary equipment for operation.

The Main Air Supply

The main air supply for the cafe is introduced through a ceiling grille at the high point and center of the ceiling with four smaller grilles in ceilings of

* Air Conditioning & Engineering Co.

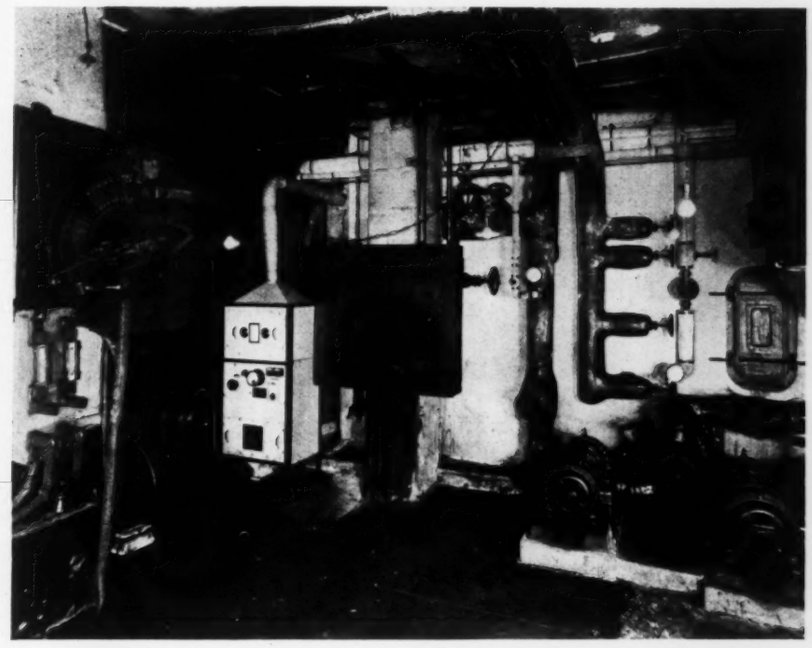


FIG. 2—FAN, WASHER AND DEHUMIDIFIER AND OZONE MACHINE

the side arches. The recirculated air is withdrawn through grilles placed at the base of four large columns which are symmetrically located near the four corners of the room. The air conditioning unit for this room has a capacity of 25,000 cfm. and is shown in Fig. 2.

The washer is furnished complete with heating sections, dampers, baffles, spray nozzles, float valve, circulating pump and control apparatus, with built-in direct expansion carbon dioxide cooling coils. Organic impurities, such as food odors, etc., are oxidized by means of ozone. The unit for the coffee shop is of similar arrangement and has a capacity of 17,500 cfm.

Temperature and Humidity Control

Each section has its own temperature and humidity differential control, operating in conjunction with a dry bulb and a dew point thermostat connected through a system of compressed air lines to a by-pass damper in the recirculating air duct of each system. When the temperatures at both thermostats rise or fall together, the thermostats neutralize each other, and a balanced condition is maintained with no effect on the by-pass damper. When the temperature at one thermostat rises or falls more rapidly than the other, an unbalanced condition is produced which causes the by-pass damper in the recirculating air duct to open or close as much as is necessary to again restore the balanced condition.

A separate exhaust system withdraws heated air from the kitchen and creates a positive air flow from dining room to kitchen.

In each system approximately 40 per cent of outside air is taken into the system from the roof of the sub-basement, where it is mixed with approximately 60 per cent of recirculated air and is drawn through the air conditioning unit, where it is cleaned, humidified or dehumidified, heated or cooled, deodorized and delivered to the room. The apparatus maintains on the hottest day in summer a room temperature of 77 F and a relative humidity of 55 per cent and is provided with ample heating sections to heat the rooms to 70 degrees in zero weather.

Saving Through Recirculation

The saving effected through the use of partial recirculation of air in this ventilating system is readily apparent, both with regard to heating and refrigerated cooling. For instance, in this system handling 25,000 cfm., practicing 75 per cent recirculation, approximately 1,250 pounds of steam per hour are saved.

Fuel Oil System For Stand-By Power Plant

THE San Francisco steam plant of the Great Western Power company of California was designed to insure continuity of service in the city of San Francisco, which is supplied, normally, by the company's hydro-electric plants through transmission lines.

In order to pick up the city load instantaneously when interruptions occur in the hydro supply, it was necessary to have a large-sized, modern steam plant, which could function much like a storage battery. The plant consists, primarily, of a 43,750 kv-a generator, driven by a 400-lb. steam turbine, supplied from two 450-lb. oil fired boilers, each of 2225 hp.

Standby operation requires each boiler to produce about 10,000 lb. of steam per hour, and an interruption in the hydro supply may require an almost instantaneous demand of steam from each of these boilers at the rate of 300,000 lb. per hour. The fuel oil burning and supply system must meet both of these extremes, and the oil must, at all times, be at the proper temperature for complete atomization and proper combustion.

To accomplish this result, a system was installed consisting of mechanical atomizing burners designed for an extremely wide range of operation, supplied by constant speed fuel oil pumps of the positive displacement type. Two sets of burners were provided on each boiler. One set consists of 9 pilots, which supply the standby requirements and serve as torches to light the second set of 20 main burners, which are used to develop maximum output of the boiler. The essential elements of the installation are shown, diagrammatically, in Fig. 1.

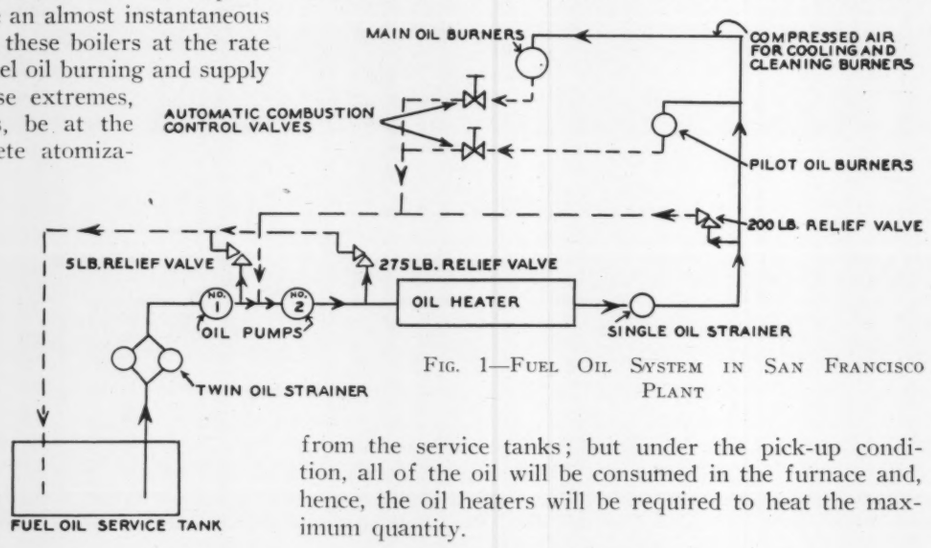


FIG. 1—FUEL OIL SYSTEM IN SAN FRANCISCO PLANT

Operation of the System

The suction of one of the fuel oil pumps is connected to the service oil tank, and its discharge to the suction of a second pump. The second pump discharges the oil through heaters and strainers to the supply headers before going to the burners. The supply headers have individual connections to the 20 main burners and 9 pilot burners on each boiler.

The oil not required for steam production is returned from the burners to a return header, which also connects to the suction of the second pump. Since the pumps discharge a constant volume, the No. 1 pump, which handles the cold oil, delivers part of its oil to the suction of No. 2 pump, and part through a low pressure relief valve to the service tank. No. 2 pump delivers most of its oil to the supply header and oil heaters, but any excess not required to maintain the supply pressure at 200 lb. is bypassed by a relief valve to the suction of this same pump, avoiding the return of hot oil to the service tank.

With this arrangement of pumps, the fuel oil system is kept hot continually and the pumps operate under a condition where the full oil requirements can be delivered to the furnace without any operating moves during the period when load is to be instantly picked up by the steam plant.

The fuel oil heaters are supplied with live steam, thermostatically controlled, to maintain constant oil temperatures and are of extra large size to heat up quickly and maintain the required temperature when the demand for oil changes suddenly from minimum to maximum.

Under the normal, or standby, condition of operation, very little oil heating is required, as the hot oil is kept in circulation and little cold oil is taken into the system from the service tanks; but under the pick-up condition, all of the oil will be consumed in the furnace and, hence, the oil heaters will be required to heat the maximum quantity.

Pumps Operate Singly or in Series

The pumps are arranged to operate in series, as outlined above, or singly, in which case each pump would draw its oil direct from the service tank and deliver directly to the supply header. This condition may obtain should the plant run for any length of time at full load, but the pumps could not be run under this condition during standby operation because of the large quantity of hot oil which would return to the service tanks and cause undue heating of them with consequent loss of suction on the pump. The series operation of pumps maintains the hot oil on the suction side of the pump under a pressure, preventing vaporization and loss of pump suction.

This system is simple to operate, and reliable, which is a requisite in a plant of this character.

The plant was designed by McClellan & Junkersfeld, Inc. H. G. Thielscher, in charge of mechanical engineering for this company, was in charge of the development of many of the new designs required to meet the unusual operating conditions of this plant. The plant was constructed by the construction department of the Great Western Power Company.—*Harold K. Fox,* Construction Engineer, Great Western Power Co.

Economical Steam Pipe Sizes
for Heating, Process and Power Steam

Article No. 2

By A. W. Moulder

With the first article of this series (which was published in the August Heating, Piping and Air Conditioning) appeared tables giving the pounds of steam carried in pipe for drops per foot in pressure of .0001 to .0080 pounds. Data to be published in following months will give this information for drops up to 1.000 pound.

The tables were explained last month; in this installment we consider a typical example, that of a heating system in an industrial building.

EXAMPLE NO. 1—INDUSTRIAL BUILDINGS

FOR this example, a diagram, Fig. 1 is shown. This is a typical small industrial heating system using five average size unit heaters, one factory radiator in the washroom and a small second floor office system at one end having four average size cast iron radiators. We will endeavor to follow through the processes by which this method of pipe sizing may be readily applied to such a system.

First step: Determine loads for each heating unit and then reduce to pounds weight of steam per hour. For such items as unit heaters, where capacities are given in Btu per hour, dividing by 1,000 Btu per pound weight of steam is close enough. Likewise, for cast iron radiation dividing amount of radiation by 4 and for pipe coils dividing amount of radiation by 3 is close enough. They can be figured closer if it is desired, but these are practical and easy factors to handle.

Making a diagram of the piping is usually unnecessary; the regular working plans ordinarily suffice. After determining the load for each heating unit, start at the far end and accumulate these loads, noting the total opposite each piece of the main supply line from tee to tee.

Mark with the load for each piece the length of that piece in feet, also the elbows or equivalent ells, if any.

It will be noted by reference to the diagram that at the extreme right at the end of the main is shown a load *G* (55 lb.) with nothing to indicate what this is for. This figure is obtained by taking 5 per cent of the entire load (1105 lb.) on the main we are considering, which is done for this reason:

Most authorities require that allowance be made for heat loss in mains by a system of figuring tentative or approximate pipe sizes, adding the amount of radiating surface of each piece and correcting the loads ac-

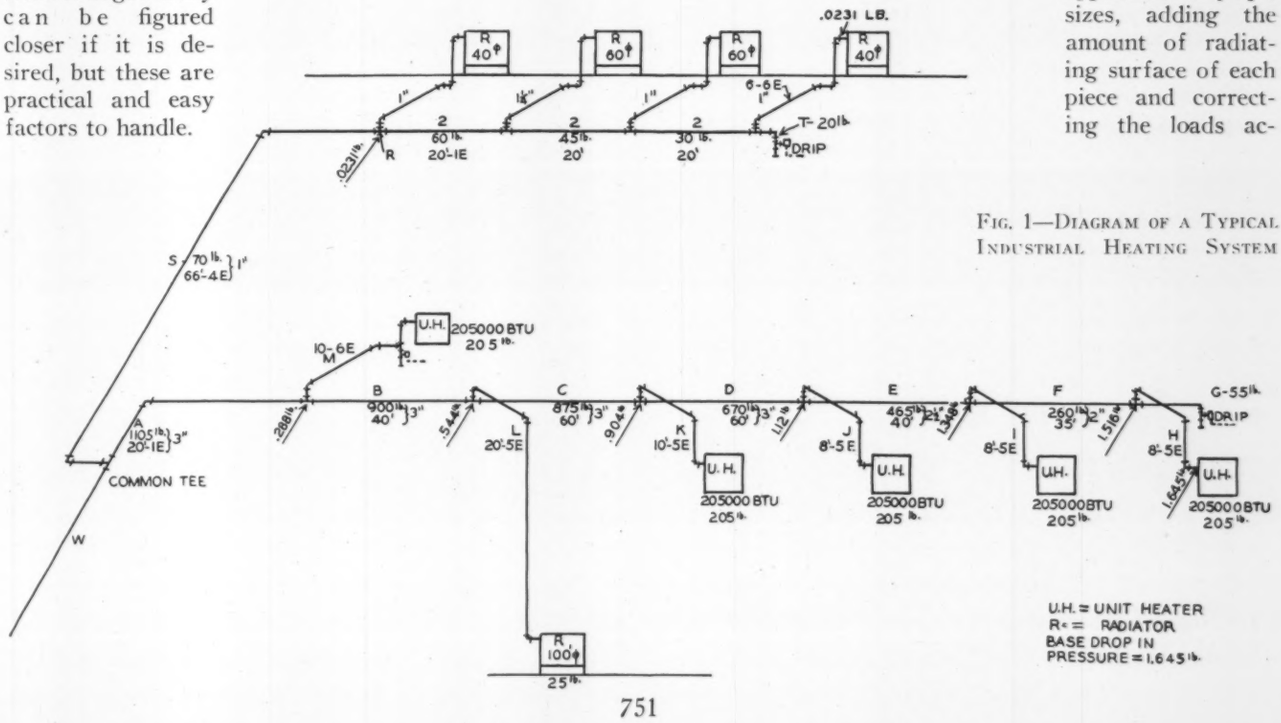


FIG. 1—DIAGRAM OF A TYPICAL INDUSTRIAL HEATING SYSTEM

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0081 to .0085 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0081	.0082	.0083	.0084	.0085
	2	3/8	5.37	5.40	5.43	5.46	5.49
	2	1/2	10.53	10.58	10.65	10.71	10.77
50′	3	3/4	24.13	24.3	24.4	24.57	24.7
	4	1	48.6	48.9	49.2	49.5	49.77
75′	5	1 1/4	106.8	107.4	108.1	108.75	109.3
100′	5	1 1/2	166	167.1	168.2	169.2	170.1
	7	2	336.5	338.5	340.6	342.6	344.6
125′	8	2 1/2	554	557.5	561	564.5	567.5
150′	10	3	1015	1021	1027	1034	1040
	12	3 1/2	1516	1525	1535	1545	1554
	14	4	2142	2155	2169	2182	2195
	15	4 1/2	2916	2934	2952	2970	2987
	17	5	3968	3992	4017	4041	4065
	20	6	6511	6551	6591	6631	6670
	24	7	9646	9704	9763	9822	9880
	27	8	13558	13641	13726	13807	13888
	30	9	18319	18431	18542.9	18654.9	18764.8
	34	10	24741.6	24892.8	25044	25195.2	25343.6
	40	12	39709.8	39952.5	40195.2	40437.8	40676.1
	47	14	51428.4	51742.7	52057	52371.3	52679.8
	53	16	74143.2	74596.3	75049.3	75502.4	75947.3
	60	18	102030.7	102654.2	103277.7	103901.3	104513.4
	67	20	135446.7	136274.5	137102.2	137929.9	138742.6
	81	24	220148.6	221494	222839.4	224184.7	225505.6

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0086 to .0090 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0086	.0087	.0088	.0089	.0090
	2	3/8	5.53	5.86	5.59	5.625	5.66
	2	1/2	10.85	10.9	10.97	11.03	11.08
50′	3	3/4	24.85	25.0	25.12	25.27	25.4
	4	1	50.05	50.3	50.6	50.9	51.25
75′	5	1 1/4	110	110.7	111.3	111.9	112.6
100′	5	1 1/2	171.1	172.2	173.1	174.1	175.1
	7	2	346.6	348.7	350.7	352.6	354.7
125′	8	2 1/2	571	574.5	578	581	584
150′	10	3	1045	1053	1057	1064	1070
	12	3 1/2	1564	1572	1581	1590	1600
	14	4	2208	2220	2232	2243	2258
	15	4 1/2	3005	3022	3043	3057	3074
	17	5	4089	4112	4135	4160	4182
	20	6	6709	6748	6786	6825	6863
	24	7	9938	9996	10053	10111	10166
	27	8	13970	14051	14131	14212	14291
	30	9	18874.7	18984.6	19092.5	19202.4	19308.3
	34	10	25492.1	25640.5	25786.2	25934.7	26077.6
	40	12	40914.4	41152.6	41386.5	41624.7	41854.2
	47	14	52988.4	53297	53599.8	53908.4	54205.5
	53	16	76392.2	76837	77273.6	77718.5	78146.9
	60	18	105125	105737.8	106338.7	106950.8	107540.4
	67	20	139555.3	140368	141165.6	141978.3	142760.9
	81	24	226826.5	228147.4	229443.8	230764.7	232036.7

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0091 to .0095 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0091	.0092	.0093	.0094	.0095
	2	3/8	5.68	5.72	5.75	5.78	5.81
	2	1/2	11.15	11.22	11.27	11.33	11.4
50′	3	3/4	25.55	25.7	25.85	26.0	26.12
75′	4	1	51.5	51.8	52.0	52.3	52.6
	5	1 1/4	113.2	113.8	114.5	115	115.7
100′	5	1 1/2	176	177	178	179	180
	7	2	356.5	358.6	360.6	362.4	364.3
125′	8	2 1/2	587.5	590.5	594	597	600
150′	10	3	1075	1082	1087	1093	1100
	12	3 1/2	1608	1616	1625	1634	1644
	14	4	2270	2282	2295	2308	2320
	15	4 1/2	3091	3108	3125	3141	3158
	17	5	4206	4229	4252	4275	4297
	20	6	6901	6939	6977	7014	7051
	24	7	10223	10279	10334	10390	10445
	27	8	14370	14449	14521	14605	14682
	30	9	19416.2	19522	19627.8	19733.7	19837.5
	34	10	26223.8	26366.3	26509.2	26652.2	26792.4
	40	12	42088	42317.4	42546.9	42776.3	43001.3
	47	14	54508.4	54805.5	55102.7	55399.8	55691.2
	53	16	78583.5	79011.9	79440.3	79868.7	80288.8
	60	18	108141.2	108730.7	109320.2	109909.7	110487.9
	67	20	143558.5	144341.1	145123.7	145906.2	146673.8
	81	24	233333.1	234605.1	235877	237149	238396.5

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0096 to .0100 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0096	.0097	.0098	.0099	.0100
	2	3/8	5.84	5.87	5.9	5.93	5.97
	2	1/2	11.46	11.5	11.57	11.63	11.7
50′	3	3/4	26.25	26.38	26.5	26.63	26.8
75′	4	1	52.9	53.2	53.45	53.7	54.0
	5	1 1/4	116.2	116.8	117.4	118.1	118.7
100′	5	1 1/2	180.8	181.7	182.6	183.6	184.5
125′	7	2	366.3	368.3	370	372	373.8
	8	2 1/2	603	606.5	609.5	612.7	615.9
150′	10	3	1105	1110	1116	1122	1127.9
	12	3 1/2	1651	1660	1668	1676	1685
	14	4	2332	2345	2357	2369	2381.7
	15	4 1/2	3175	3191	3208	3224	3240.7
	17	5	4320	4342	4365	4386	4409.1
	20	6	7089	7125	7162	7198	7234.9
	24	7	10501	10554	10609	10663	10717.3
	27	8	14761	14836	14913	14988	15064.9
	30	9	19943.3	20045.1	20148.9	20250.7	20354.5
	34	10	26935.3	27072.8	27213	27350.4	27490.7
	40	12	43230.8	43451.4	43676.4	43897	44122
	47	14	55988.4	56274.1	56565.5	56851.2	57142.7
	53	16	80717.2	81129.1	81549.2	81961.1	82381.3
	60	18	111077.4	111644.3	112222.4	112789.3	113367.5
	67	20	147456.4	148208.8	148976.4	149728.9	150496.4
	81	24	239668.5	240891.5	242139.1	243362.1	244609.6

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Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0102 to .0110 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0102	.0104	.0106	.0108	.0110
	2	⅜	6.02	6.08	6.14	6.2	6.26
50′	2	½	11.8	11.92	12.04	12.15	12.27
	3	¾	26.85	27.15	27.4	27.66	27.9
75′	4	1	54.5	55.0	55.6	56.1	56.6
	5	1¼	119.8	120.8	122.3	123.4	124.5
100′	5	1½	186.3	188.2	190	191.8	193.5
125′	7	2	377.4	381	385	388.7	392
150′	8	2½	621.5	628	634	640	646
	10	3	1138	1150	1161	1172	1182
	12	3½	1700	1716	1732	1750	1766
	14	4	2402	2429	2452	2475	2500
	15	4½	3272	3304	3336	3367	3398
	17	5	4453	4496	4539	4582	4624
	20	6	7307	7378	7448	7519	7588
	24	7	10823	10930	11033	11137	11240
	27	8	15214	15363	15509	15656	15800
	30	9	20556	20757.5	20955	21152.4	21347.8
	34	10	27762.8	28035	28301.6	28568.3	28832.2
	40	12	44558.8	44995.6	45423.6	45851.6	46275.2
	47	14	57708.4	58274.1	58828.4	59382.7	59931.2
	53	16	83196.9	84012.4	84811.5	85610.6	86401.5
	60	18	114489.8	115612.1	116711.8	117811.5	118899.8
	67	20	151986.3	153476.2	154936	156395.9	157840.6
	81	24	247031.2	249452.9	251825.6	254198.3	256546.6

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0112 to .0120 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0112	.0114	.0116	.0118	.0120
	2	⅜	6.315	6.37	6.43	6.48	6.54
50′	2	½	12.38	12.5	12.59	12.7	12.82
	3	¾	28.17	28.4	28.67	28.9	29.15
75′	4	1	57.17	57.6	58.2	58.65	59.15
	5	1¼	125.7	126.8	127.9	129	130.1
100′	5	1½	195.3	197	198.8	200.4	202.3
125′	7	2	395.7	399.2	403	406	409.5
150′	8	2½	652	657	663.5	669	674
	10	3	1194	1204	1215	1225	1235
	12	3½	1783	1798	1815	1831	1845
	14	4	2520	2540	2562	2585	2603
	15	4½	3429	3460	3493	3520	3549
	17	5	4666	4708	4749	4789	4830
	20	6	7657	7725	7792	7859	7925
	24	7	11342	11443	11543	11641	11740
	27	8	15943	16085	16225	16364	16502
	30	9	21541.2	21732.5	21921.8	22109.1	22296.3
	34	10	29093.4	29351.8	29607.4	29860.4	30113.3
	40	12	46694.3	47109.3	47519.4	47925.3	48331.3
	47	14	60474.1	61011.2	61542.7	62068.4	62594.1
	53	16	87184.1	87958.5	88724.7	89482.6	90240.5
	60	18	119976.	121042.4	122096.7	123139.7	124182.7
	67	20	159270.3	160685	162084.6	163469.2	164853.8
	81	24	258870.4	261169.7	263444.6	265695	267945.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0122 to .0130 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0122	.0124	.0126	.0128	.0130
	2	⅜	6.575	6.65	6.7	6.75	6.815
50′	2	½	12.93	13.03	13.13	13.23	13.34
	3	¾	29.4	29.6	29.85	30.1	30.57
75′	4	1	59.65	60.0	60.6	61.1	61.6
	5	1¼	131.2	132.2	133.3	134.3	135.4
100′	5	1½	203.8	205.7	207.3	208.7	210.5
125′	7	2	413	416.4	420	423	426.5
150′	8	2½	680	686	691	696.5	702
	10	3	1245	1255	1265	1275	1285
	12	3½	1860	1875	1890	1905	1920
	14	4	2628	2650	2672	2690	2715
	15	4½	3579	3608	3637	3666	3694
	17	5	4869	4909	4949	4988	5027
	20	6	7991	8056	8121	8185	8249
	24	7	11837	11934	12030	12125	12219
	27	8	16639	16775	16910	17043	17176
	30	9	22481.6	22664.7	22847.9	23027.1	23206.2
	34	10	30363.4	30610.8	30858.3	31100.2	31342.1
	40	12	48732.8	49129.9	49527	49915.3	50303.5
	47	14	63114.1	63628.4	64142.7	64645.5	65148.4
	53	16	90990.1	91731.6	92473	93198	93922.9
	60	18	125214.4	126234.7	127255	128252.6	129250.2
	67	20	166223.3	167577.7	168932.2	170256.6	171580.9
	81	24	270171.3	272372.8	274574.3	276726.9	278879.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0132 to .0140 Lb.

Velocity Ft. Per. Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0132	.0134	.0136	.0138	.0140
	2	⅜	6.85	6.91	6.96	7.02	7.07
50′	2	½	13.44	13.54	13.65	13.75	13.85
	3	¾	30.78	31.03	31.26	31.47	31.72
75′	4	1	62.0	62.5	62.95	63.4	63.9
	5	1¼	136.5	137.5	138.5	139.5	140.5
100′	5	1½	212	213.6	215.2	216.7	218.4
125′	7	2	429.6	433	436	439	442.5
150′	8	2½	707	713	718	723	729
	10	3	1297	1305	1315	1325	1335
	12	3½	1935	1950	1965	1980	1995
	14	4	2735	2755	2775	2798	2818
	15	4½	3723	3751	3779	3806	3834
	17	5	5066	5103	5141	5179	5217
	20	6	8312	8374	8437	8499	8560
	24	7	12313	12405	12497	12590	12681
	27	8	17308	17438	17567	17697	17825
	30	9	23385.3	23560.3	23735.4	23910.4	24083.5
	34	10	31584	31820.4	32056.9	32293.3	32526.9
	40	12	50691.8	51071.3	51450.7	51830.2	52205.2
	47	14	65651.2	66142.6	66634.1	67125.5	67611.2
	53	16	94647.9	95356.3	96064.8	96773.3	97473.6
	60	18	130247.9	131222.8	132197.8	133172.8	134136.4
	67	20	172905.3	174199.6	175493.8	176788.1	178067.3
	81	24	281032	283135.6	285239.3	287342.9	289422.1

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Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0142 to .0150 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0142	.0144	.0146	.0148	.0150
	2	3/8	7.115	7.17	7.21	7.26	7.315
50′	2	1/2	13.93	14.05	14.13	14.23	14.33
	3	3/4	31.95	32.17	32.4	32.63	32.76
75′	4	1	64.3	64.8	65.3	65.6	66.15
100′	5	1 1/4	141.5	142.5	143.5	144.5	145.4
125′	5	1 1/2	220	221.6	223	224.4	226
150′	7	2	445.5	448.7	451.7	455	457.7
	8	2 1/2	734	739	744	749	754
	10	3	1345	1353	1362	1372	1382
	12	3 1/2	2010	2022	2036	2050	2065
	14	4	2838	2858	2878	2898	2912
	15	4 1/2	3861	3880	3915	3942	3968
	17	5	5254	5291	5328	5364	5400
	20	6	8621	8682	8742	8801	8861
	24	7	12771	12861	12950	13038	13125
	27	8	17951	18078	18203	18327	18450
	30	9	24254.4	24425.4	24594.4	24761.3	24928.2
	34	10	32757.9	32988.8	33217	33442.4	33667.8
	40	12	52575.8	52946.4	53312.7	53674.5	54036.3
	47	14	68091.2	68571.2	69045.5	69514.1	69982.6
	53	16	98165.6	98857.6	99541.3	100216.8	100892.4
	60	18	135088.7	136041	136981.9	137911.5	138841.1
	67	20	179331.5	180595.7	181844.8	183078.9	184312.9
	81	24	291476.8	293531.5	295561.8	297567.6	299573.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0152 to .0160 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0152	.0154	.0156	.0158	.0160
	2	3/8	7.36	7.41	7.45	7.49	7.55
50′	2	1/2	14.42	14.52	14.62	14.7	14.8
75′	3	3/4	33.05	33.26	33.5	33.67	33.9
100′	4	1	66.55	67.0	67.5	67.8	68.3
125′	5	1 1/4	146.4	147.4	148.3	149.3	150.1
150′	5	1 1/2	227.5	229	230.6	232	233.3
	7	2	461	464	467	470	472.5
	8	2 1/2	759	764	769	774	779
	10	3	1390	1398	1410	1418	1426
	12	3 1/2	2080	2090	2106	2118	2131
	14	4	2935	2952	2975	2993	3010
	15	4 1/2	3995	4021	4047	4073	4099
	17	5	5436	5471	5507	5542	5577
	20	6	8919	8978	9036	9094	9152
	24	7	13212	13299	13386	13471	13556
	27	8	18572	18694	18816	18935	19056
	30	9	25093	25257.9	25422.8	25583.6	25746.4
	34	10	33890.5	34113.2	34335.8	34553	34772.9
	40	12	54393.6	54751	55108.4	55457	55810
	47	14	70445.5	70908.4	71371.2	71822.6	72279.8
	53	16	101559.7	102227	102894.2	103545.1	104204.1
	60	18	139759.4	140677.7	141596	142491.6	143398.5
	67	20	185532	186751	187970	189158.9	190362.9
	81	24	301554.7	303536.1	305517.4	307449.8	309406.7

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0162 to .0170 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0162	.0164	.0166	.0168	.0170
	2	3/8	7.60	7.65	7.685	7.73	7.78
50′	2	1/2	14.88	14.99	15.07	15.16	15.23
75′	3	3/4	34.13	34.33	34.54	34.74	34.94
100′	4	1	68.7	69.15	69.55	70	70.4
125′	5	1 1/4	151.1	152.1	153	153.8	154.8
150′	5	1 1/2	235	236.3	237.8	239.3	240.5
	7	2	476	478.7	482	484.6	487.4
	8	2 1/2	783.5	788.5	793.5	798	803
	10	3	1435	1444	1453	1462	1470
	12	3 1/2	2145	2159	2170	2185	2195
	14	4	3030	3048	3068	3085	3100
	15	4 1/2	4124	4150	4175	4200	4225
	17	5	5611	5646	5681	5715	5749
	20	6	9208	9265	9322	9377	9433
	24	7	13640	13725	13808	13891	13973
	27	8	19173	19292	19410	19526	19642
	30	9	25905.2	26066	26224.8	26381.5	26538.2
	34	10	34987.4	35204.5	35419	35630.6	35842.3
	40	12	56154.1	56502.7	56846.8	57186.6	57526.3
	47	14	72725.5	73176.9	73622.6	74062.6	74502.6
	53	16	104846.7	105497.5	106140.1	106774.4	107408.7
	60	18	144282.8	145178.4	146062.6	146935.6	147808.5
	67	20	191536.8	192725.7	193899.6	195058.4	196217.2
	81	24	311314.7	313247.1	315155	317038.5	318922

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0172 to .0180 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0172	.0174	.0176	.0178	.0180
	2	3/8	7.83	7.87	7.915	7.97	8.015
50′	2	1/2	15.35	15.43	15.5	15.61	15.7
75′	3	3/4	35.14	35.37	35.55	35.77	35.96
100′	4	1	70.8	71.3	71.6	72.0	72.4
125′	5	1 1/4	155.7	156.7	157.5	158.4	159.3
150′	5	1 1/2	242	243.5	244.8	246.2	247.6
	7	2	490.3	493.3	496	499	501.5
	8	2 1/2	808	812.5	817	822	826
	10	3	1480	1489	1495	1504	1513
	12	3 1/2	2210	2220	2235	2247	2260
	14	4	3120	3140	3160	3175	3194
	15	4 1/2	4249	4274	4299	4323	4347
	17	5	5782	5816	5849	5882	5915
	20	6	9488	9543	9598	9652	9706
	24	7	14055	14136	14218	14298	14378
	27	8	19756	19871	19985	20098	20211
	30	9	26692.9	26847.6	27002.3	27155	27307.6
	34	10	36051.3	36260.2	36469.1	36675.3	36881.5
	40	12	57861.6	58197	58532.3	58863.2	59194.1
	47	14	74936.9	75371.2	75805.5	76234.1	76662.6
	53	16	108034.8	108660.9	109287	109904.9	110522.8
	60	18	148670.1	149531.7	150393.3	151243.5	152093.8
	67	20	197361	198504.8	199648.5	200777.3	201906
	81	24	320781.1	322640.1	324499.1	326333.7	328168.3

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Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0182 to .0190 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0182	.0184	.0186	.0188	.0190
50′	2	⅜	8.05	8.09	8.14	8.18	8.225
	2	½	15.78	15.87	15.95	16.05	16.12
75′	3	¾	36.17	36.35	36.55	36.75	36.95
100′	4	1	72.9	73.2	73.6	74.0	74.4
	5	1¼	160.2	161	161.9	162.8	163.7
125′	5	1½	249	250.3	251.7	253	254.3
150′	7	2	504.5	507	510	512.5	515
	8	2½	830	835	840	844	849
	10	3	1522	1530	1536	1546	1555
	12	3½	2275	2285	2298	2310	2322
	14	4	3213	3230	3245	3262	3280
	15	4½	4371	4395	4419	4443	4467
	17	5	5948	5980	6013	6045	6077
	20	6	9760	9814	9867	9920	9973
	24	7	14458	14537	14616	14695	14773
	27	8	20323	20434	20546	20656	20766
	30	9	27458.2	27608.9	27759.5	27908.1	28056.7
	34	10	37084.9	37288.3	37491.8	37692.5	37893.1
	40	12	59520.6	59847.1	60173.6	60495.7	60817.8
	47	14	77085.5	77508.3	77931.2	78348.3	78765.5
	53	16	111132.4	111742	112351.6	112953	113554.4
	60	18	152932.7	153771.6	154610.5	155438.1	156265.7
	67	20	203019.6	204133.3	205247	206345.6	207444.2
	81	24	329978.4	331788.5	333598.6	335384.3	337169.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0192 to .0200 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0192	.0194	.0196	.0198	.0200
50′	2	⅜	8.275	8.315	8.36	8.4	8.45
	2	½	16.20	16.28	16.39	16.47	16.57
75′	3	¾	37.14	37.34	37.54	37.74	37.91
100′	4	1	74.8	75.2	75.6	76.0	76.4
125′	5	1¼	164.5	165.4	166.3	167	167.9
150′	5	1½	255.7	257	258.5	259.7	261
	7	2	518	520.8	523.5	526	528.8
	8	2½	853	857.5	862.5	867	871
	10	3	1562	1570	1579	1586	1594
	12	3½	2333	2345	2360	2370	2380
	14	4	3300	3318	3332	3350	3368
	15	4½	4490	4513	4537	4560	4583
	17	5	6109	6141	6173	6204	6235
	20	6	10024	10077	10129	10180	10232
	24	7	14850	14927	15004	15080	15156
	27	8	20874	20982	21091	21198	21305
	30	9	28203.2	28349.8	28496.3	28640.8	28787.4
	34	10	38091.1	38289	38486.9	38682.1	38877.3
	40	12	61135.5	61453.2	61770.9	62084.1	62397.4
	47	14	79176.9	79588.3	79999.8	80405.5	80811.2
	53	16	114147.5	114740.7	115333.8	115918.7	116503.6
	60	18	157082	157898.2	158714.5	159519.4	160324.3
	67	20	208527.8	209611.4	210695	211763.5	212832
	81	24	338931.1	340692.3	342453.5	344190.2	345926.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0205 to .0225 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0205	.0210	.0215	.0220	.0225
50′	2	⅜	8.55	8.65	8.75	8.85	8.95
	2	½	16.76	16.96	17.15	17.36	17.56
75′	3	¾	38.38	38.86	39.3	39.76	40.23
100′	4	1	77.25	78.25	79.2	80.1	81.0
125′	5	1¼	170	172	174	176.2	178
150′	5	1½	264.3	267.5	270.6	273.8	277
	7	2	535.5	542	548	554.5	561
	8	2½	882	893	903	914	924
	10	3	1614	1635	1654	1672	1692
	12	3½	2415	2440	2470	2500	2528
	14	4	3410.7	3451.2	3491.6	3532.1	3572.6
	15	4½	4640	4695	4751	4806	4861.1
	17	5	6314	6389	6464	6539	6614
	20	6	10360	10483	10606	10729	10852
	24	7	15347	15529	15712	15894	16076
	27	8	21573	21829	22085	22341	22597
	30	9	29147.7	29493.7	29839.7	30185.8	30531.8
	34	10	39366.6	39834	40301.3	40768.7	41236
	40	12	63182.8	63932.8	64682.9	65433	66183.1
	47	14	81828.3	82799.8	83771.2	84742.6	85714
	53	16	117970	119370.5	120771	122171.5	123572
	60	18	162342.2	164269.5	166196.7	168124	170051.2
	67	20	215510.8	218069.3	220627.7	223186.2	225744.6
	81	24	350281	354439.3	358597.7	362756.1	366914.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0230 to .0250 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0230	.0235	.0240	.0245	.0250
50′	2	⅜	9.05	9.15	9.24	9.34	9.435
75′	2	½	17.73	17.93	18.13	18.32	18.5
100′	3	¾	40.65	41.1	41.54	41.96	42.4
125′	4	1	81.9	82.8	83.7	84.5	85.4
150′	5	1¼	180	182	184	185.8	187.7
	5	1½	280	283	286	289	292
	7	2	567	573.5	579	585	591
	8	2½	934	944	954	964	974
	10	3	1710	1730	1748	1765	1783
	12	3½	2555	2579	2610	2635	2662
	14	4	3610.7	3651.2	3689.3	3727.4	3765.5
	15	4½	4912.9	4968.1	5019.9	5071.7	5123.6
	17	5	6684	6759	6830	6900	6971
	20	6	10968	11091	11207	11323	11439
	24	7	16247	16430	16601	16773	16944
	27	8	22838	23095	23336	23577	23818
	30	9	30857.5	31203.5	31529.2	31854.8	32180.5
	34	10	41675.9	42143.2	42583	43022.9	43462.7
	40	12	66889	67639.1	68345	69051	69757
	47	14	86628.3	87599.7	88514	89428.3	90342.6
	53	16	124890.1	126290.5	127608.6	128926.7	130244.8
	60	18	171865.1	173792.3	175606.2	177420.1	179234
	67	20	228152.5	230711	233118.9	235526.9	237934.8
	81	24	370828.2	374986.5	378900.3	382814.1	386727.8

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cordingly. From the standpoint of operating efficiency, the principal purpose of calculating pipe sizes rather accurately either by this method or any other is to maintain a practically uniform steam pressure at the farthest unit just the same as at the units nearest the source of supply. In order to aid in this, the main piping is usually insulated when, as a matter of fact, it is perfectly good and valuable heating suface. The value of main pipe heating surface is particularly enhanced in the modern industrial or mercantile buildings, many of which are heated with overhead type unit heaters, which tend to pull overheated air from the ceilings and discharge it downward. Therefore, in this instance we want to figure the main piping as not insulated.

The amount of heating surface in the form of exposed piping in a heating system may be placed between 5 and 10 per cent of the radiation or equivalent radiation. In this case, as with most unit heater systems, the proportion of piping to the equivalent radiation (merely by judgment on the part of the engineer) is very low, and we therefore place it at 5 per cent.

From a purely practical standpoint (disregarding theory entirely) we have found that it is good policy to make this allowance at the far end of the main pipe. This means deliberately making a small error in favor of the isolated heating units. A compensating factor is that a considerable amount of moisture is entrained as the main increases in length. Much of this entrained moisture may be carried into the smaller piping at the end in spite of all possible provisions for proper dripping. Therefore, in this example we have added approximately 5 per cent or 55 lb. to the load at the end of the mains.

It is usually convenient to accumulate the loads and lengths taken from the plans in something like the following form:

Piece Mark	Load, Pounds	Length		Equivalent Length
		Ft.	Ells	
A	1105	20	1	30
B	900	40	..	40
C	875	60	..	60
D	670	60	..	60
E	465	40	..	40
F	260	35	..	35
G	....	..	..	..
H	205	8	5	43
		263	6	308

Second step: Having tabulated the dimensions of the various parts of the piping with the load on same, the second step is to determine the sizes of this piping. In order to do this, however, we first come to the question of the total drop in pressure which may be used up in the system. In deciding this, the engineer must consider several practical aspects:

(a) If the steam supply is from a low pressure boiler, with direct gravity return, the head from the lowest heating unit to the water line should govern. In such cases, if possible, a factor of safety of 2 or 3 to 1 should be allowed.

(b) If exhaust steam is used, the maximum allowable back pressure must be considered.

(c) If steam is drawn through a pressure reducing valve, only practical and economical pipe sizes are used.

An engineer with experience soon learns to choose a suitable total drop in pressure for the system. In this example we will assume an approximate total allowable drop of 2 lb.

We find by adding our total lengths and assuming an average size of 2 in. for the ells that we have an approximate total equivalent length of 300 ft. Dividing this into 2 lb., we get an average theoretical drop of .0066 lb. per foot.

We continue with this second step, therefore, by picking the sizes of pipe which must closely approximate the average drop of .0066 and then calculating the actual drop for each piece, accumulating the total drop at each tee where a branch connection is taken off. Such calculations, when tabulated and the loads and lengths explained under first step, appear in the following table:

For Permanent Record

Piece Mark	Load, Pounds	Length		Equiv. Length	Size, Inches	Drop per Ft.	Drop in Piece	Total Drop
		Ft.	Ells					
A	1105	20	1	30	3	.0096	.288	.288
B	900	40	..	40	3	.0064	.256	.544
C	875	60	..	60	3	.0060	.360	.904
D	670	60	..	60	3	.0036	.216	1.120
E	465	40	..	40	2½	.0057	.228	1.348
F	260	35	..	35	2	.0048	.168	1.516
G	....	..	..	..	..	.....	....	.....
H	205	8	5	43	2	.0030	.129	1.645
		263	6	308				
		1st Step			2nd Step			

A choice between two pipe sizes for a given load often causes a variation of 50 per cent above or below the average to be maintained. For example, piece *A*, 3½ in. pipe has a drop of .0046 per foot, whereas 3 in. is .0096 per foot. It is usually best to start with a drop higher than average so as to cut down the sizes of the larger pipes and then use sizes which show drops below average at the far end of the mains where the sizes are much smaller. Obviously by so doing the saving in first cost is more pronounced.

In calculating the size of pipe for the last connection, in this case *H*, it is also well to figure a size of pipe which will be liberal in size even though somewhat below the average estimated drop. Usually the same size of connection that the engineer would ordinarily use for the same unit or radiator if using a rule-of-thumb method of determination will be about right for this connection.

The reasons for this are that little is saved in first cost by making this farthest connection small in size, and equalization of other connections is much easier if this is not reduced too much, all of which will be shown later in these calculations.

Third step: At this point it will be seen that we have arrived at a total theoretical drop in pressure from the first or "common" tee (so-called because it is the starting junction, back of which any loss is common to every branch) to the end of the farthest connection of 1.645 lb. This is called the *base drop*. In the perfect system, the drop in pressure to the end of every connec-

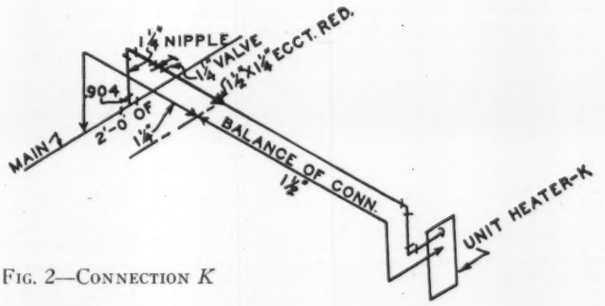

FIG. 2—CONNECTION *K*

tion, i.e., the entrance to every radiator or unit on the job, should be 1.645 lb.

It is possible to accomplish this to an entirely practical degree. For the purpose of illustration we will use connection *K* (see Fig. 2) as an example. It is suggested that calculations such as the following (or a simplification of same) for every connection be kept as a permanent record.

Connection *K* (Fig. 2)

Load: 205 lb.

10 ft. and 5 ells

Drop in main at connection: .904 lb.

Base drop: 1.645 lb. — .904 lb. = .741 required drop through connection.

Assuming connection size of 1½" the equivalent length would be 35 ft.

.741 ÷ 35 ft. = .021 average drop per ft. on basis of connection being all 1½"

Referring to Fig. 2A for a drop of .021 we find the 1½" connection would pass 267.5 lb. of steam or 62.5 lb. more than required. This indicates the load of

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0205 to .0225 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot .0205	.0210	.0215	.0220	.0225
	2	¾	8.55	8.65	8.75	8.85	
	5	1¼		78.25	79.2	80.1	81.0
100′	5	1¼	170	172	174	176.2	178
125′	5	1½	264.3	267.5	270.6	273.8	277
150′	7	2	535.5	542	548	554.5	561
	8	2½	882	893	903	914	924
	10	3	1614	1635	1654	1673	
	12			2440			

FIG. 2A

205 lb. comes between 1¼" and 1½". In this case the amount of 1¼" pipe to install is determined as follows:

205 lb. on 1¼" pipe = .0300 lb. per ft. (See Fig. 2B.)

205 lb. on 1½" pipe = .0124 lb. per ft. (See Fig. 2C.)

.0176 difference in drop per ft.

.0124 × 35 ft. = .434 lb. drop if connection was all 1½" pipe

.741 required drop — .434 = .307 lb. drop to be added in connection by use of 1¼" pipe

.307 ÷ .0176 = 17 equivalent ft. of 1¼" pipe required, the balance of run to be 1½" pipe.

As for applying this result to the drawings, it is

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0280 to .0300 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot .0280	.0285	.0290	.0295	.0300
	2	⅜	9.99	10.07	10.15	10.23	10.33
75′	2	½	19.57	19.75	19.9	20.1	20.27
100′	3	¾	44.7	45.26	45.66	46.05	46.44
125′	4	1	90.4	91.2	92.0	92.8	93.6
150′	5	1¼	198.6	200.5	202.2	203.9	205.7
	5	1½	308.8	311.6	314.4	316	319.8
	7	2		631	637	642	648

FIG. 2B

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0122 to .0130 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot .0122	.0124	.0126	.0128	.0130
	2	⅜	6.575	6.65	6.7	6.75	6.815
50′	2	½	12.93	13.03	13.13		
	3						
	5	1¼	131.2		133.3	134.3	
100′	5	1½	203.8	205.7	207.3	208.7	210.5
125′	7	2	413	416.4	420	423	426.5
				686	691		702

FIG. 2C

usually necessary in such a case to make a small diagrammatic sketch of the connection to show how properly to apply same, similar to Fig. 2.

In arriving at the foregoing we have figured:

Main outlet tee	= (2 Ells) =	10′ Equiv. length
Riser nipple	=	1′ Equiv. length
Ell	=	5′ Equiv. length
Pipe to reducer	=	1′ Equiv. length
	Total	17′ Equiv. length 1¼"

Attention is directed to the fact that in this sketch the exact equivalent length of 1¼ in. pipe is figured out and shown, whereas it is simply noted that the remainder of the connection is 1½ in. Allowance is made for the fact that no matter how carefully the layout is made, the length of connections will vary somewhat and therefore it is better that this "more or less" be taken up in the 1½ in. size where the drop in pressure per ft. will affect the total drop in the connection relatively less than the 1¼ in. size. The same calculation should be made for connections *I*, *J* and *L*, Fig. 1.

The Balance Between the Practical and the Theoretically Perfect

Some engineers do not consider it practical to go so far as to make reductions in the piping connections to obtain this high degree of equalization. As before stated, it is the purpose to show just how far it has been found (in actual practice over a long time and great volume of work) really practical to go toward approaching the theoretically perfect in steam pipe sizing. Therefore, we must state that for industrial heating piping of the type used in the example it has really

proven practical to go as far as shown. Of course, this assumes that a fairly accurate layout is being made as to location of main piping and location of heating units.

The importance of going into as much detail as shown varies, of course, somewhat with the type of system. If a vacuum system with traps and steam is supplied through a pressure reducing valve, then the importance is somewhat lessened because we know that the equalization in the return piping is relatively unimportant (though still worth while) and that ample steam pressure to fill all units to maximum capacity will be always available. On the other hand, if steam is taken from a low pressure boiler with more or less intermittent supply and variable pressures, the careful equalization (getting steam to and maintaining steam in those farthest

10 ft. and 6 ells
Drop in main at connection: .288
Base drop: 1.645 lb. — .288 lb. = 1.357 required drop through connection.
Assumed 2-in., equivalent length 52 ft.
205 lb. on 1″ pipe = .140 lb. drop per ft. (See Fig. 3A.)
205 lb. on 2″ pipe = .0030 lb. drop per ft. (See Fig. 3B.)

.1370 difference in drop per ft.
.0030 × 52 ft. = .1560 drop if connection was all 2″ pipe.
1.357 required drop — .1560 = 1.201 lb. drop to be added in connection by use of 1″ pipe.
1.201 ÷ .1370 = 8 equivalent ft. of 1″ pipe required, the balance of run to be 2″ pipe.

Connection *M* (Fig. 3) has been taken deliberately as

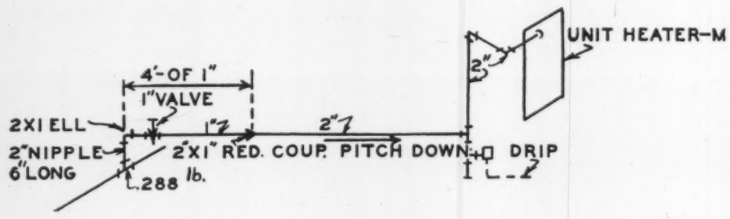

FIG. 3—CONNECTION *M*, A UNIT HEATER HIGHER THAN THE MAINS

units as well as the units nearest the source of supply) is obviously a worthwhile object.

We are presenting this aspect of the problem only as we have found it practical, but have no argument with the engineer who, for reasons, insists that he wants to make the connection *K* (Fig. 2) in the foregoing example all 1½ in. instead of "going the limit." However, we submit that even this reduction to 1½ in., where it would ordinarily be at least 2 in., is an improvement.

Fourth step: In all of the foregoing, we have assumed examples where velocity in the piping did not enter the problem at all. In other words, we could ignore velocity because, as stated once before, this need only be considered where condensate must be carried back against the flow of steam.

To show what can be done toward equalization in cases where some such problem must be met we have saved connection *M* (Fig. 3) as our example. We are assuming in this case that the unit heater is considerably higher than the mains, and that a connection must be run upwards to supply it.

Fig. 3 is a diagrammatic sketch of this connection.

The explanation is as follows:

Connection *M* (Fig. 3)

Load—205 lb.

an extreme example to show just how far it is practical to go with this. As to the velocity consideration, the 2 in. size was chosen because the velocity for the vertical connection upward is then only slightly over 50 ft. per second, well within the range of allowable velocities. In fact, velocities up to 150 ft. per second, while possibly impractical for long vertical risers, are found entirely satisfactory for short vertical risers which carry only a very nominal amount of condensation.

Because of the strategic location of unit *M*, in that it is placed so close to the source of supply of steam, the required throttling effect as shown in the calculations forces us to use 1 in. pipe for that purpose. When we get into such extreme conditions, it is doubly important that extreme care be used in locating the length and position of the throttling piece. It will be noted that, in this case, the reducing ell at the top of riser nipple has been figured the same as a 1 in. ell or 1 in. outlet of a slightly larger tee which is the equivalent of 4 ft. of 1 in. pipe and to which has been added 4 ft. of actual 1 in. pipe. A throttle piece, if judgment is used by the engineer, may always be placed in a manner which will assure reasonable accuracy of installation.

FIG. 3A

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .1250 to .2250 Lb

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.1250	.1500	.1750	.2000	.2250
125′	2	¾	21.09	23.1	24.95	26.7	28.3
150′	2	1	41.35	45.3	49.0	52.3	55.5
	3	1¼	94.8	103.8	112.1	119.85	127.15
	4	1½	190.8	209.2	225.7	241.5	256.2
	5	2	420	460			563

FIG. 3B

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0026 to .0030 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0026	.0027	.0028	.0029	.0030
25′	2	¾	3.035	3.095	3.155	3.21	3.26
	2	1	5.96	6.08	6.19		6.41
50′	5	1½	94.1	95.9	97.7	99.4	101.1
	7	2	190.5	194.3	197.8	201.3	204.7
75′	8	2½	314	320	326	331.5	337
	10	3	566.1	586	597	608	617.5
	12	3½	859	875	892	907	922
			1215				

For connection *M* (Fig. 3), if any engineer should choose not to equalize too closely but only approximately (in this line with the 1½ in. size for connection *K* (Fig. 2)), then a 1¼ in. connection taken off from the main and extending to bottom of 2 in. riser piece would represent good judgment.

Fifth step: The steam pipe sizing work for the principal part of the system has been completed in the preceding four steps of Example No. 1 (Fig. 1), but there remains to determine the size of piping for the branch line to the four average size cast iron radiators located in the factory offices.

It is quite impractical and usually not worth the effort to calculate and endeavor to graduate the pipe sizes for each individual radiator in this small branch system. The logical and practical solution, therefore, is to treat the entire group of radiators from "common tee" *R* (upper part of Fig. 1) to the end of the main as one unit.

This permits calculating the sizes for these radiator connections and branch main from "common tee" *R* on to the end by any rule-of-thumb basis that the engineer knows from experience will produce satisfactory and noiseless results for the type of system involved. We will assume that this section is to be a one-pipe system and will apply ordinary pipe sizes in common usage accordingly. These sizes are shown on Fig. 1.

Obviously the drop in pressure in the piping for this unit of four radiators is negligible, and the engineer, experienced in this method of pipe sizing, would immediately recognize this fact and ignore this drop as a factor.

However, in order to prove that this is a fact and because we are working in this instance with reasonable accuracy we will figure the actual resistance.

UNIT *R*

LOAD	SIZE	LENGTH	DROP PER FT.	DROP IN PIECE	TOTAL DROP
60 lb.	2 in.	20′ + 7′ = 27′	.0003	.0081	.0081
45 lb.	2 in.	20′	.0002	.0040	.0121
30 lb.	2 in.	20′	.0001	.0020	.0141
10 lb.	1 in.	6′ + 24′ = 30′	.0003	.0090	.0231

In other words, with ordinary commercial pipe sizes for this part of the system, the total drop in pressure from point *R* to the end of the system is .0231 lb., which, in proportion to the base drop of 1.645 lb., might easily be neglected. Obviously, also, the question of equalizing the connection to each individual radiator may be neglected.

In any event, if we calculate the pressure drop to point *R* through connection *S* at approximately the same as the base drop of 1.645 lb., then this part of the system will have been equalized with the remainder of the system. The calculations therefore follow:

Connection *S*

Load—70 lb.

66′—4 Ells

Drop in Unit *R*—.0231

Base drop 1.645 — .0231 = 1.6219 required drop through connection

Drop— 1″ pipe .0172 per ft.

Drop—¾″ pipe .0690 per ft.

Pounds of Steam Carried in Pipe for Drops per Lineal Foot of .0660 to .0700 Lb.

VELOCITY FT. PER SEC.	EQUIV. LENGTHS IN FEET OF EACH ELBOW	PIPE SIZE	POUNDS DROP PER LINEAL FOOT				
			.0660	.0670	.0680	.0690	.0700
100′	2	¾	15.33	15.45	15.55	15.67	15.77
125′	2	1	30.05	30.3	30.45	30.7	30.95
150′	3	¾	68.9	69.4	69.9	70.4	70.92
	4	1	138.75	139.8	140.8	141.8	142.85

FIG. 3C—DROP IN THE ¾ IN. PIPE IS .0690 WITH A 70-LB. LOAD (WITH FIGURES AS CLOSE AS 69.9 AND 70.4, THE HIGHER VALUE IS USED)

On basis of all 1″:

66′ + 16′ = 82′ equiv. length = 1.41

After arriving at a drop in pressure for this connection as close as that shown by calculating all 1 in. pipe, it would be obviously foolish to figure further with a view to equalizing exactly. Furthermore, it is usually well to favor this type of connection slightly so as to be certain that the office section gets at least pressure equal with the balance of the system.

Final step: This completes the calculations for our Example No. 1, with the exception only of the size of piping back of the common tee where the previous calculations started.

What happens by way of calculating pressure drop in the line back of this point is governed entirely by the source of supply. In other words, there will be an economical size of pipe to be determined dependent entirely upon how much additional total pressure may be lost through same. By figuring at most 2 or 3 sizes of pipe for this connection, the practical size of pipe will usually be obvious to the engineer.

Editor's Note: In the October issue of HEATING, PIPING AND AIR CONDITIONING the application of this method of calculating steam pipe sizes will be shown for heating systems characteristic of offices, hotels, etc.

Committee to Standardize Gages

Upon the request of the *American Standards Association*, the *American Society of Mechanical Engineers* recently accepted sole sponsorship for the development of an American standard for pressure and vacuum gages. In order that this sectional committee may meet early in the fall, invitations have been addressed to fifty-five societies and associations requesting them to appoint representatives on this committee. The tentative scope outlined for this project is: "Nomenclature and definitions of pressure and vacuum gages; capacity ratings; case sizes and mounting holes with a view to obtaining maximum interchangeability; dials and graduations and designation of units; indicator hand and bushing; zero stop pins, bezel rings and their attachment to the gage; connections to the gage; method of expressing allowable errors or accuracy of the gage; requirements for accuracy in so far as establishment of such requirements proves to be feasible; methods of testing; rules and specifications for installation and use of pressure and vacuum gages."

Air Conditioning a Box Shop

By R. S. Hawley*

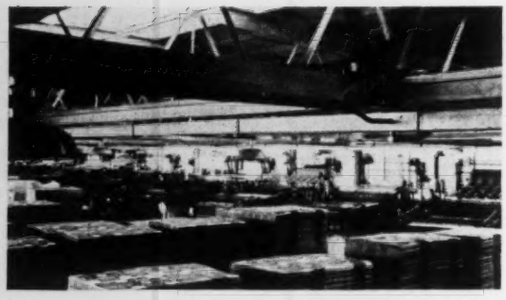

FIG. 1—INSTALLATION OF A MODERN AIR CONDITIONING SYSTEM WAS, OF COURSE, PART OF THE EXPANSION PROGRAM OF THE CONSOLIDATED PAPER COMPANY AT MONROE, MICH.

DURING the past two years the Consolidated Paper Company, whose main offices and principal plants are located at Monroe, Michigan, have included in their expansion program the erection of a large building and the installation of the necessary equipment to make it a completely modern factory for the manufacture of paper board containers.

The building itself is approximately 400 feet in length by 100 feet in width and is four stories high, exclusive of the basement. The top floor is devoted exclusively to the printing department. Fig. 1 shows a view of a battery of printing presses on this floor. The cutting and forming equipment occupies most of the third floor. The second floor is used primarily for temporary storage of finished board and the first floor is devoted to weighing, transfer, and shipping except for a mill-wright shop which is located at the south end of this room. The basement is partitioned and is used partly for garage purposes and partly for storage.

Air Conditioning System Installed

In addition to the equipment which has been installed for the manufacture of containers, one of the outstanding features of the factory equipment is the air conditioning system which was installed after a considerable amount of investigation and study. It is the purpose of this article to tell something of the problem involved, the type of equipment installed and the results obtained.

The importance of maintaining a correct uniform temperature in printing establishments is, of course, well known to heating engineers. The consistency of ink varies rapidly with changes of temperature and the difficulties of proper impressions are almost directly proportional to the variations in temperature, particularly if the temperature is allowed to drop below 70 to 72 F. Automatic temperature control is therefore an essential to maintain the uniformity necessary.

Humidity Control Important

Scarcely less important than temperature control in a container box shop is the value of a proper degree of humidity in the atmosphere. When board is manufactured it contains a certain amount of moisture. When left in large rolls or bales in a cool storage room and not subject to air changes, this moisture will be retained almost constant for long periods of time. But when the individual sheet is exposed to the atmosphere it will absorb or reject moisture very rapidly. This causes either expansion or shrinkage which results in poor impressions or incorrect size of container, and if the moisture content of the board is too low, increased breakage will result. For these reasons humidity and humidity control were given serious consideration and were included in the air conditioning system for this plant.

Dust a Problem

A third problem which the foreman of a box shop has to deal with is that of dust and dirt. When container board is cut into necessary shapes for finished boxes a large quantity of paper dust is caused, some of which is very fine and light and almost invisible. In addition to this there are the clippings which are, however, much less of a problem to take care of than the fine dust. Particles of dust

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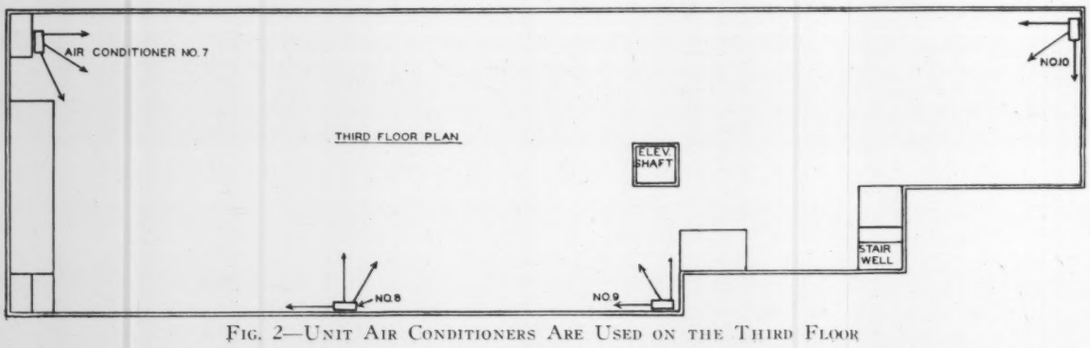


FIG. 2—UNIT AIR CONDITIONERS ARE USED ON THE THIRD FLOOR

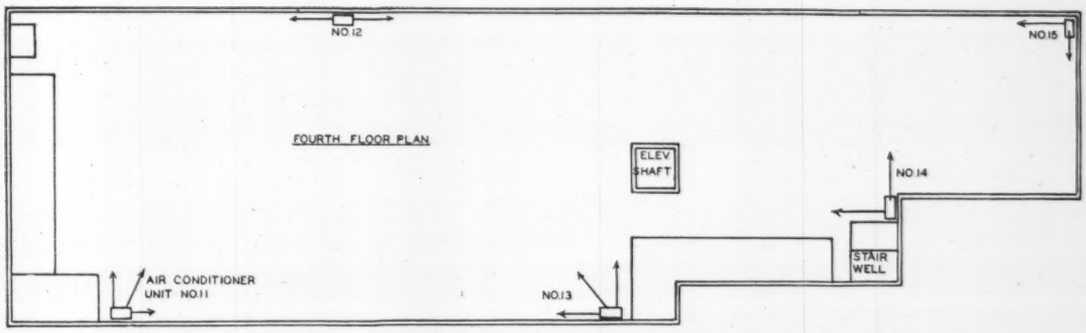


FIG. 2A—LOCATION OF THE AIR CONDITIONING UNITS ON THE FOURTH FLOOR

in the air cause trouble in printing, mar the appearance of the container and increase the wastage. Air washing was the third point to be considered.

The ultimate aim in the selection of air conditioning equipment for this plant was not the most perfect system that could be designed and installed but rather the system which would operate within the limits considered necessary for this type of plant and would produce the desired results with the lowest ultimate cost.

It was decided to install unit air-conditioners on the three upper floors and unit heaters on the first floor. Five air conditioning units were installed on the top floor, four on the third floor and three on the second floor. Small rooms are heated by direct radiation. Figs. 2 and 2A show the location plan of the units on the two upper floors and Fig. 3 is a view of one of the units. Each unit consists of a tank at the base with two vertical-shaft motor-driven pumps and sprays. Above this is an eliminator, next the heating coils and at the top is a motor-driven fan and diffusion duct. Air passes down into the base of the machine, then upward across the spray, the heating coils and the fan and is discharged horizontally into the room at a height to clear the stacks of board and the room equipment. There is one combined thermostat and humidostat for each unit, and these are located at points which were selected to maintain in the room as uniform a condition as possible. Each thermostat functions by admitting or releasing compressed air to a diaphragm valve at each unit while the humidostat controls a second valve which admits steam to the water in the washer tank. The relative humidity is therefore governed by the temperature of the water in the tank.

It will be evident that close control of humidity will not easily be secured by this type of apparatus. The system is re-circulating and while humidity may be added to the air it is difficult to de-humidify. However, it was thought that control would be close enough for the purposes. The humidostats are set to close with a relative humidity of 50 to 52 per cent and a series of readings taken over a considerable period of time indicate that the relative humidity does not vary over 5 per cent and this appears to be close enough for practical purposes in this plant. Temperatures of 68 to 72 F are easily and uniformily maintained on the two upper floors while on the first and second floor the temperature is carried several degrees lower.

The washer elements have proven to be very effective in the removal of dust, dirt and fiber from the air. Some trouble has been experienced in the pump bearings but on the whole the units have given satisfactory operation without an excessive amount of attention or maintenance. The contracting firm of Johnson, Larsen and Co. of Detroit, installed the equipment and did the piping work.

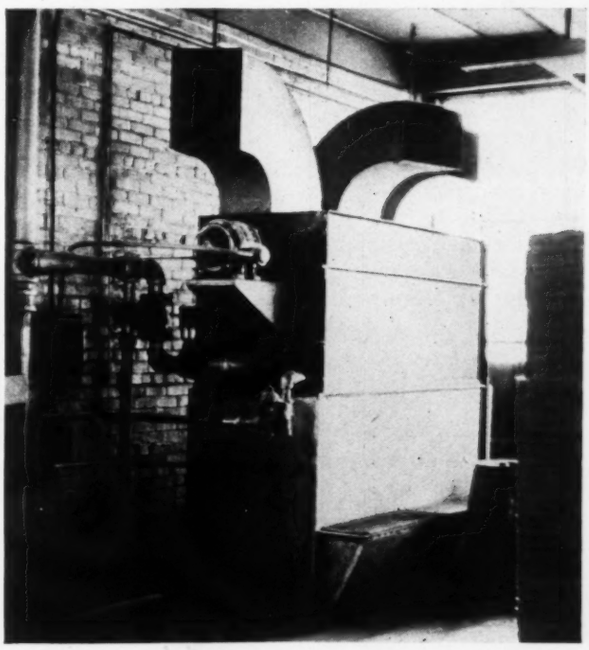


FIG. 3—A VIEW OF ONE OF THE AIR CONDITIONING UNITS INSTALLED IN A FACTORY FOR THE MANUFACTURE OF PAPER BOARD CONTAINERS

Condensate from

By R. M. McQuitty*

TEN STORY OFFICE BUILDING OF THE UNION ELECTRIC LIGHT AND POWER COMPANY, IN ST. LOUIS, IN WHICH IS INSTALLED THE CONDENSATE RECOVERY SYSTEM. LARGE SAVINGS HAVE BEEN EFFECTED SINCE ITS INSTALLATION LESS THAN A YEAR AGO

WHY should a building throw away pure, soft, hot water and then pay for heating city water for building use? This is exactly what is being done in many buildings which are purchasing steam for heating purposes. This practice has been established due to the extensive growth of central station (district) steam heating. These buildings have found it more profitable to purchase steam than to generate it in their own boiler plants. The central station has found it more and more uneconomical to return the condensate to the plant, with the result that today the steam is often purchased by the customer, and the condensate discharged to the sewer on the premises.

Approximately 11 Per Cent of Original Btu Value of Steam Contained in Condensate

Each pound of steam purchased (assumed at 5 lb. gage pressure, dry saturated) contains approximately 1,114 Btu. After the heat of vaporization is given up in the heating systems the steam becomes condensate and is discharged from the heating system returns at a temperature of approximately 160 F. This hot condensate still contains approximately 11.2 per cent of the original heat value of the steam.

Condensate Suitable for Hot Water Supply

The condensate that drains from the heating system is, in the majority of cases, pure distilled water, which could not be excelled by the most expensive softening and purifying treatment. It is usually hot enough to be used as hot water without being reheated, and the amount of condensate available during the heating season is usually very nearly the same as the requirements for hot water.

Test Installation Made in Company Office Building

In order that a complete study might be made as to the advisability of using condensate as hot water, a condensate recovery system was installed, as shown in the accompanying photograph, in the ten-story office building of the Union Electric Light and Power Company at St. Louis, Missouri. This installation is equipped with special metering and temperature recording equipment and the piping is arranged so that methods of operation may be varied.

The first consideration in this study was purity of the condensate. A chemical analysis was made of both steam and condensate. This analysis showed the condensate here to be exceptionally pure, containing only .048 grains of total solids per gallon. No trace of boiler compound could be detected, in either the steam or condensate. At the time the analysis was made a special study was made as to the corrosive effects the condensate might have on hot water piping. The conclusion reached was that the corrosive effect would be negligible, due to the general tendency for the condensate to be mixed with the city water.

Actual practice has demonstrated that the coating formed in the pipes due to use of hard water tends to counteract any corrosive effect of the condensate. The condensate, in turn, practically removes the usual trouble due to pipes becoming stopped up from coating deposits.

Design of Condensate Recovery System

Fig. 1 shows the design of the recovery system. This system has the special feature whereby the condensate can be either used as it is available or stored for future use. It is entirely automatic in operation. If the supply of condensate is insufficient to supply the hot water demand, cold water is heated by a thermostatically controlled steam valve. If the supply of condensate is greater than the demand for hot water, the condensate is stored for future use.

The details of operation of the system are as follows:

The condensate is collected as it drains from the heating returns in the float tank, from which it is automatically pumped into the hot water heater. An equal volume of water is forced out of the hot water heater through the hot water piping, back through the hot water returns and into the top of the storage tank. As the storage tank is being filled with hot water, cold water, which remains at the bottom, is forced out and backed up into the cold water house tank. When the storage tank becomes filled with hot water, the water being backed out becomes warm and causes a temperature control switch to stop the pump. Any excess condensate then fills the float tank and overflows to the sewer until

* Union Electric Light & Power Co., St. Louis, Mo.

Heating System Used as Hot Water Supply for Building

some of the hot water is used. If the supply of condensate is insufficient, then the storage tank is emptied, after which the cold water from the house tank fills the storage tank and is supplied to the hot water heater. This lowers the temperature in the hot water storage heater and steam is admitted to the coils by the thermostatically controlled steam valve.

The use of this system also gives the building the benefit of forced circulation in the hot water piping. A lower temperature can thus be maintained in the hot water heater to supply a given average temperature throughout the hot water system.

Temperatures Maintained

The temperature of the condensate at this building varies from 100 degrees to 180 degrees fahrenheit. However, the extreme variations occur only when the system is first heated. The usual temperature of the condensate is approximately 150 degrees.

The temperature of the condensate will vary considerably with respect to different types of heating systems. Many of the modern systems are able to cool it as low as 100 degrees, while the one-pipe gravity systems often discharge condensate as high as 200 degrees.

The thermostatically controlled steam valve is set to open if the water in the heater falls below 130 degrees. This seems to be the best point for condensate temperatures averaging 150 degrees. Provided the supply of condensate is sufficient to meet the hot water demand, a fairly constant temperature of 140 degrees can be maintained in the hot water supply with 150 degree condensate. The extreme variations do not affect the temperature of the hot water supply to any great extent, due to the large amount of storage.

Danger from high temperatures in the hot water supply has been negligible in the above installation. However, this can be guarded against by installing another temperature control switch in the hot water heater. This switch should be wired in series with the other control switch and the float switch. Excessive temperatures would then cause the pump to stop, and cold water would be mixed with the hot condensate.

Savings Effected

The recovery system was first put in operation in the building December 16, 1929; therefore, a complete year's data is not available at this time. The savings accomplished so far, as compared to the previous year, are shown in Table 1 on the next page.

Application to Various Types of Buildings

The recovery system is applicable to any type of building having need for hot water, provided the condensate is suitable for hot water supply.

The design as shown in Fig. 1 is applicable to any building which has a cold water house tank. In this design the pump should have sufficient capacity to reclaim the maximum amount of condensate which will be available at any time. This is especially necessary during the warmer months of the heating season when all condensate available can be used.

The amount of storage capacity required for various other installations will, of course, depend upon the nature of the hot water demand, and the condensate available. Many buildings are already equipped with a spare hot water heater. One of the heaters can then be used as a storage tank as well.

Fig. 2 shows another design of recovery system which is applicable to buildings not having a cold water house tank, or, it may be used in any case where the hot water load is fairly constant.

In this method the storage capacity required is ac-

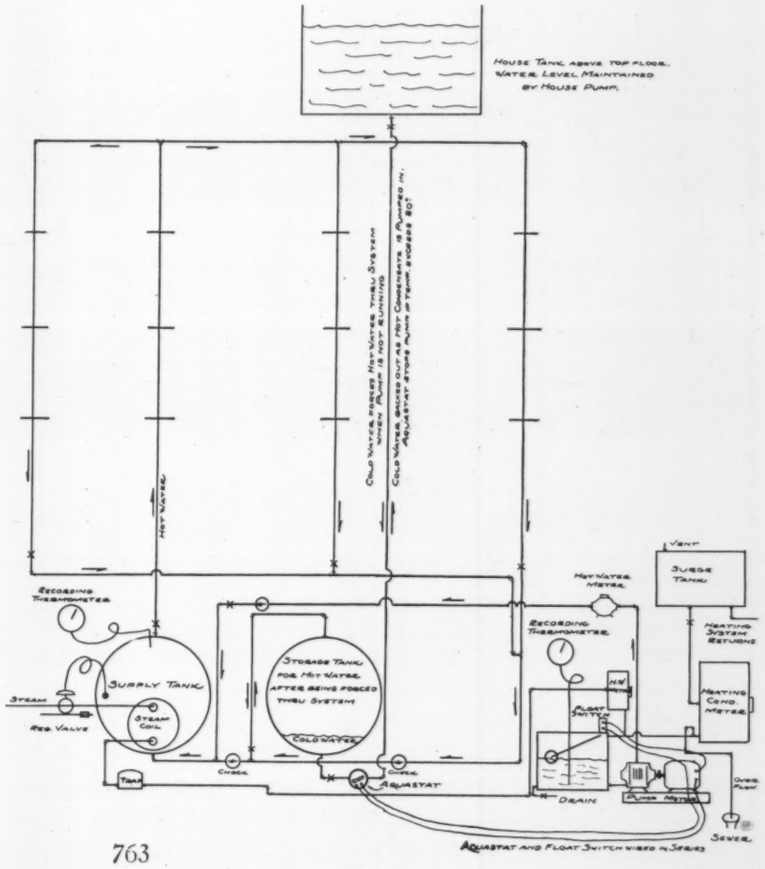


FIG. 1—CONDENSATE RECOVERY SYSTEM. DESIGN FOR FORCED CIRCULATION

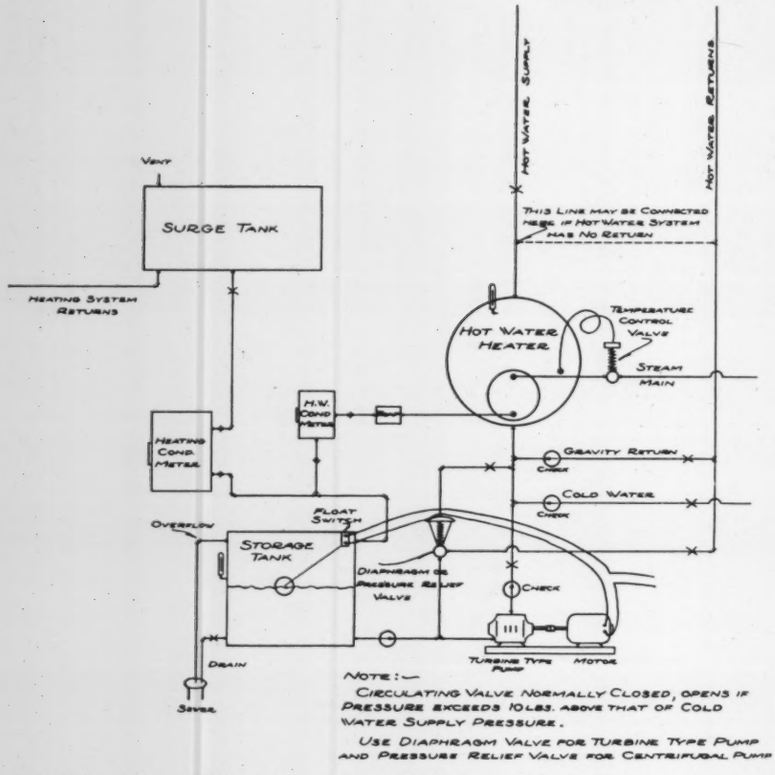


FIG. 2—CONDENSATE RECOVERY SYSTEM. SPECIAL DESIGN FOR BUILDINGS HAVING NO COLD WATER HOUSE TANK OR BUILDINGS HAVING A FAIRLY CONSTANT HOT WATER DEMAND

complished by making the condensate float tank extra large. The condensate is pumped into the hot water heater at a constant rate as long as it is available. If the hot water demand is less than the amount being pumped in, the circulating valve is opened by the higher pressure in the system and the pump merely circulates the water in the system. Although this method requires that the pump be operating regardless of the hot water demand, as long as there is condensate available, the amount of electric current consumed by the motor is at a minimum during the time the pump is merely circulating the water in the system.

The type of circulating valve required will depend upon the type of pump used. For the comparatively low static head, the ordinary centrifugal type pump can be used. This type of pump has a characteristic whereby the horse power required is indirectly proportional to the static head. Thus, when no water is being drawn from the system and the pressure builds up this type of pump requires very little power. However, the pump cannot be allowed to churn, as the friction would finally heat the water in the pump casing, and the pump might be ruined. In order, then, for this type of pump to be operating at the maximum head, a pressure relief valve should be installed so that the pump will circulate just enough water to keep it from overheating on no load. The pressure relief valve would then replace the diaphragm valve shown in Fig. 2.

The turbine type pump, which is often used for the higher pressures, has a characteristic as to power and static head which is opposite to the centrifugal pump. The power required for this pump increases with the pressure. It is desirable, then, for this type of pump to operate against a minimum effective static head during the time when it is simply circulating the water in the system. This is accomplished with the diaphragm type valve which opens when the pressure increases. The pressure on the inlet to the pump is thus increased which reduces the effective static head.

Methods of Estimation of Savings Where Condensate Is Utilized

Since the amount of condensate available varies with the heating requirements, while hot water demand remains more or less constant during any month of the year, any estimate of savings should be made on a monthly basis.

The steam required to heat a given quantity of water per month, a part of which is supplied by condensate, may be calculated for all practical purposes from the following formula:

W = 1,000 pounds hot water required per month.

C = 1,000 pounds condensate reclaimed per month from the building heating system (assumed to be as hot as the required hot water temperature).

K = a constant = pounds water which can be heated

TABLE 1

COMPARATIVE STEAM CONSUMPTION FOR WATER HEATING WITH AND WITHOUT CONDENSATE RECOVERY SYSTEM

Cubical contents of Building ..... 1,307,158 cu. ft.
Radiation connected ..... 32,041 sq. ft.
Hot water requirements 2000 to 3000 gallons per day.
Size of pump used to return condensate ..... 12 G.P.M.
Storage capacity ..... 300 gallons

Month	M. Lb. Steam Required				D.D.D.'s (Based on 70F)		M. Pounds Hot Water Required	Cond. Available	Cond. Reclaimed as H. W.
	Bldg. Heating		Water Heating						
	1929	1930	1929	1930	1929	1930			
Jan.	1,425	1,603	72	0	1,326	1,385	737	1,603	737
Feb.	1,207	880	77	17	1,165	698	693	897	465
Mar.	646	997	79	8	622	807	711	1,005	592
April	249	317	67	43	370	318	603	360	278
May	176	10	73	49	298	163	524	59	59
June	12	.....	55	42	.....	.....	462	42	42
July	.....	.....	52	38	.....	.....	418	38	38
			(1928)	(1929)					
Dec. (15	days)		79	20	.....	.....	355	699	341
Total	3,715	3,807	554	217	3,781	3,371	4,503	4,703	2,552

Saving in Steam for above period ..... 337,000 pounds
Saving in water approximately 2,552,000 lbs. or ..... 36,618 cu. ft.
Percentage saving in Steam 60.8%
Percentage of Total Condensate available reclaimed 54.3%
Saving in cost of Steam and Water based on St. Louis rates $252.00
Cost of Recovery system Installed 425.00
Total Estimated saving per year 350.00

by 1 pound of steam where condensate is not returned.

Steam required per month for water heating, if condensate $= \frac{W - C}{K + 1}$ M. pounds of steam is utilized.

Where water is heated from 60 degrees to 160 degrees Fahrenheit with steam at 5 pounds gage pressure (dry saturated), K becomes approximately 9.

The value of K may be calculated for other conditions as follows:

$$K = \frac{Q_s - Q_c X}{Q_{w2} - Q_{w1}}$$ (per cent efficiency of heater)

Where,

Q_s = Btu content of 1 pound of steam.

Q_c = Btu content of 1 pound of condensate as discharged from heater.

Q_{w1} = Btu content of 1 pound water to be heated.

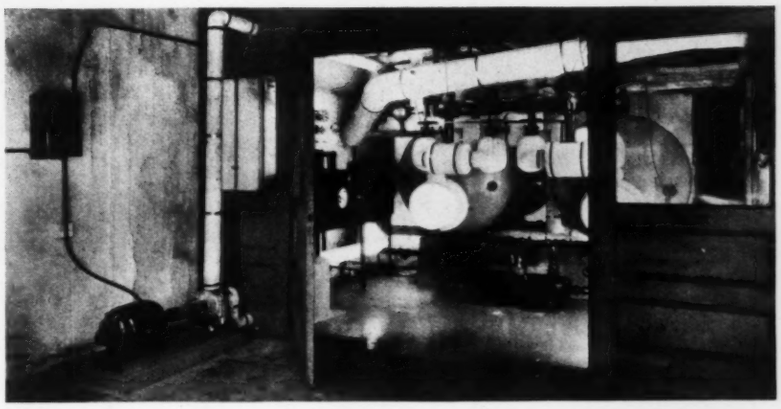
Q_{w2} = Btu content of 1 pound water after being heated.

For all practical purposes K can be assumed as equal to 9 and the formula becomes:

M. pound steam required =

[(M pounds water required) — (M pounds condensate from heating system) per month] ÷ 10

A study of ten office buildings in St. Louis indicates that the annual hot water requirements are approximately equal to the condensate available. However, the condensate supply usually exceeds the demand for hot water during the coldest months of the season, and a considerable amount of condensate is therefore lost. By applying the above formula for savings to these buildings, an office building to have normal heating and hot water requirements should be able to reclaim approximately 62 per cent of all condensate available during the year.



THE CONDENSATE RECOVERY SYSTEM INSTALLED IN THE BASEMENT OF THE UNION ELECTRIC LIGHT AND POWER COMPANY OFFICE BUILDING

Hotels require from three to five times as much hot water as office buildings. An estimate based on one hotel indicates that approximately 90 per cent of the condensate available could be utilized.

The study or the advisability of using condensate as hot water supply is of course limited to those buildings served by the district heating system of the Union Electric Light and Power Company, so it would not necessarily mean that the condensate from all systems would be suitable. However, a chemical analysis can always be made which will definitely indicate whether or not the condensate in any locality is adaptable as hot water supply. Even though the condensate is unsuitable for hot water, it can still be utilized by means of the economizer. A possible saving of 11 per cent in steam consumption is certainly worth consideration of some method of utilization.

Elaborate System of Sound Insulation in Newspaper Building

HEATING and air conditioning engineers are, of course, familiar with the insulation of building construction to prevent the flow of heat into or out of structures. Piping engineers, too, come in daily contact with materials to insulate against heat.

The acoustical engineer encounters similar problems in preventing the transmission of sound through buildings. Insulation for sound isolation is very important in numerous buildings; especially so in structures containing heavy machinery in one part and offices in another.

One of the most elaborate systems of sound insulation ever installed protects the Daily News building in Chicago from noise and vibration in the pressroom. This gigantic room, three stories in height, houses presses comprising three banks of forty-five units and requiring 1,800 hp. daily for operation. The aggregate weight of the presses and folders amounts to some 1,440 tons. This fast-running machinery produces an unusual amount of vibration and noise, which architects and engineers succeeded in eliminating from the rest of the building by ingenious means.

Floor Slab of Concrete

The floor slab of the pressroom, covering an area of approximately 20,000 square feet, is of concrete from eight to ten inches thick. The recesses in the floor immediately underneath each unit of the presses are lined with a three-inch layer of cork cut in rectangles 24 inches wide and 36 inches long, bound together with steel straps. Above the cork is a layer of building paper mopped with tar, which prevents water from another concrete slab above from penetrating into the cork. This final layer of re-enforced concrete is eighteen inches thick. Upon it are set the presses.

Vibration Not Communicated to Walls

Practically all of the vibration and much of the noise is eliminated by this process. The vibration is absorbed by the cork cushion and not communicated to or through the floor, ceiling, walls and pillars. Expert engineers declare that the job is unusually successful in the insulation of sound and vibration. The insulation is doubly requisite in the building, since virtually two-thirds of the structure is leased for private offices.

Special Ceiling

To reduce further the noise from the pressroom, architects specified a special ceiling, of perforated metal and one inch of insulation. With vibration insulated by the cork layer in the floor recesses and around the bases of each column, and noise reduced to a minimum by the cushion and the special ceiling, the system as a whole undoubtedly represents a notable installation of insulation for the isolation of sound.—*J. V. D.*

Life Sciences Building

In the recently completed Life Sciences building, located on the campus of the University of California at Berkeley, a vault twenty-nine feet square is used for the storage of skins and furs. Its thermostat maintains a temperature of 38 degress Fahrenheit, the variation being no greater than one degree either way. This is said to be an ideal temperature for the storage of valuable furs and skins and will insure their preservation.

Another vault containing bones and skeletons is of the same size, but its temperature is automatically maintained at 55 degrees. This temperature is best-suited for the indefinite preservation of bony structures.

The building is of concrete construction, and is fire- and earthquake-proof. Five stories high, it cost two million dollars to erect.

Removing Moisture From Air by Adsorption

THE adsorption process is often used in place of cooling to remove moisture from air. This involves the use of any chemical which has the power to adsorb large quantities of water and which will give it up readily on the application of heat.

Many chemicals have, of course, such characteristics, but other requirements such as low cost, speed of adsorption and desorption over a practical range of temperatures, and ease of handling naturally narrow the field considerably. One chemical group which has won much favor for this purpose is the gels.

These gels are placed in metal containers installed in air ducts in such a manner that the air is compelled to pass around the gel crystals. The resistance of the air through the container is determined by the tightness of the pack of the gel and the mesh size of the crystals used. Where continuous operation is necessary, two or more containers must be employed so that the gel crystals in one container can be dehydrated while another container holding gel is in service.

The characteristic performance curve of one of these gels is given in Fig. 1 for an air temperature of 86 F and an air flow of 3 cu. in. per min. through 10 grains weight of the gel. Saturation for this gel was reached when it had absorbed one-quarter of its weight in water.

It should be noted that the efficiency of adsorption falls off at an extremely rapid rate from the end of the second hour. At the same time the weight of the gel is increased at an equally rapid pace by the water which is taken up. A point is thus quickly reached where it does not pay to continue the adsorption process longer with the gel in use. In order to maintain a high overall efficiency for the adsorption process it is thus necessary to discontinue the use of the gel after it has taken up less than 30 per cent of its weight in water. In Fig. 1 the time interval for such a period is less than two and a quarter hours. In practice this interval can be made four hours by increasing the amount of gel per container. The next step is to "activate" the gel or to get rid of the water which the gel has absorbed. This water is driven off by switching the container out of the air line, by means of blast gate valves, and heating the gel crystals to a temperature of about 400 F for a predetermined period of time.

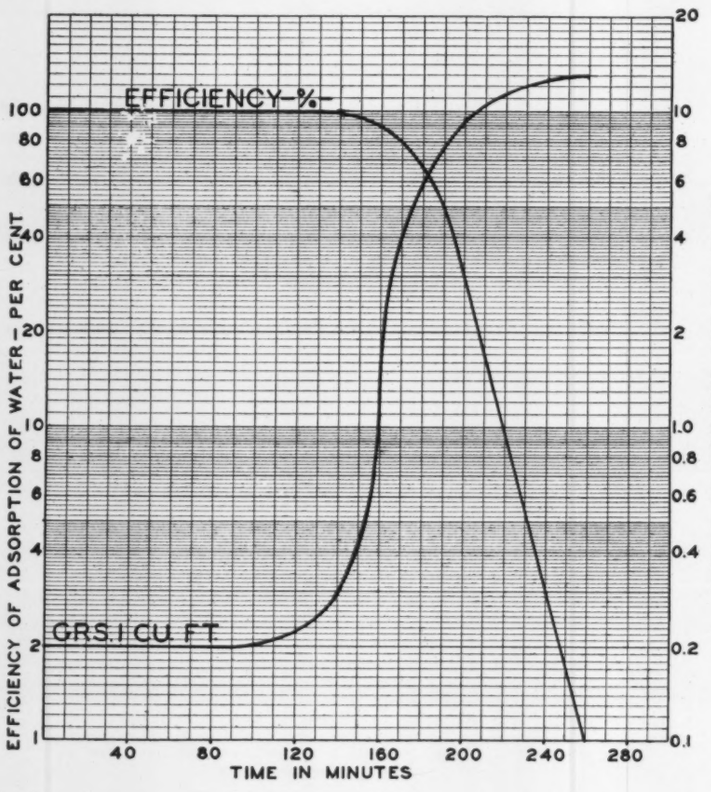


FIG. 1.—PERFORMANCE CURVES FOR ADSORPTION OF WATER BY A GEL

Low Rate of Flow

The example given in Fig. 1 is for a very low rate of air flow. Rates comparable with usual air conditioning needs can be had without difficulty. For example, 400 cu. ft. per min. air flow, for a dewpoint of —30 F or a relative humidity of ½ per cent for a dry bulb temperature of 110 F required about 600 pounds of gel per container where the period in the air line did not exceed 8 hours. In this particular case, the air introduced into the gel had a dry bulb temperature of 110 F. and a relative humidity of 10 per cent. —*Malcolm Tomlinson.*

Unit Heaters Prevent Condensation in Dye Houses

By John R. Cooper*

To the plant engineer who was confronted with moisture and foggy conditions in the buildings under his charge during the cold months last winter, this article will point the way to a simple method of eliminating this expensive and dangerous problem. Now is the time to prepare industrial plants for the coming winter—unit heaters as installed in the typical cases given here may prove the answer.

THE problem of the removal of moisture always exists whenever we have steaming tubs throwing off moisture in confined places and is, of course, most objectionable, as well as dangerous when outside air temperatures are low; in other words, during the winter months.

The reasons are simple: There are definite limits to the amount of moisture that air can absorb, depending upon its temperature. When moisture to an excess of that amount is given off by operating conditions we pass the dew point of the air and immediately have fog conditions.

When this foggy air comes into contact with relatively cold surfaces, such as roof trusses, skylights and iron framework, exposed to outside lower temperatures, we immediately have condensation problems that are not only annoying to the working force but also expensive as they cause corrosion to all metal parts of roof construction and also damage to goods in the way of causing spots that entail a duplication of work already done, to remove.

Since the operating temperature of dye houses is approximately 75-80 F and the moisture is being continually thrown off by the tubs, the air, unless changed frequently, will soon reach its absorptive limit and, when we add the constant cold air infiltration through exterior walls and windows, the fog conditions often reach a point where it is almost impossible for the operators to see what they are doing or where they are going.

This is a decidedly dangerous situation which it is necessary to correct.

When fog conditions reach this point, the first thought of the owner (unless familiar with the problem) is to remove the fog as though it were smoke, by exhausting it, and he is then considerably surprised to find that his condition is worse than ever, in spite of the large amount of fog that is obviously being removed.

He forgets that simple basic fact, that Nature won't stand for a vacuum and in consequence, the air which is being removed, in the form of fog, must be replaced by a similar amount pulled, in most cases, from either colder rooms or by infiltration through exterior walls and windows.

This colder air lowers the room temperature, reduces the dew point and the room simply changes from a warm, foggy condition to a colder, still more foggy one, in spite of all efforts and expense.

About that time, he calls for help and gets his problem solved more or less satisfactorily, dependent upon where he goes for proper information.

Is Problem of Air Transportation

As a matter of fact, the problem is an absurdly simple one if properly analyzed, consisting as it does of nothing more than a question of *air transportation.*

If air at a certain temperature won't hold more than a certain amount of moisture and more than that amount is being continually furnished, that must be taken care of, the logical answer is to furnish the proper amount of additional air required for the purpose.

This air must, of course, be heated up to at least room temperature prior to admission, or we have the same effect as previously discussed.

Since zero air will absorb only about 1/16 the amount of moisture that 70 deg. air will, it is obvious that if we heat up outside air, at about to 75-80 deg., we certainly have an exceedingly "hungry bunch of air" that we can devote to our purpose.

We say "hungry," advisedly, in view of the marked affinity air has for moisture, under such conditions.

The only question is how much and where.

Experience tells us that conditions naturally vary and the number of air changes that will be required will be somewhere between 6 and 16 per hour.

Necessary Air Changes

As a rule, 10 air changes per hour will be sufficient, if exhaust fans are used to remove the saturated air, and about 15 per hour if the air removal is caused by the pressure built up by this continual volume of air being forced into the room from outside.

The use of exhaust fans and the proper location of unit heaters to insure as great a uniformity of air circulation as possible will, unquestionably, tend to prevent condensate problems, as the quicker the air movement over cold spots the less opportunity to condense the moisture into grief-making drops.

Boiled down, the problem consists of calculating the

* Thermal Units Co.

cubic contents of the room, multiply it by the number of air changes decided upon and then install the proper number of units to give the necessary volumes at the proper temperature.

Since volumes are the keynote the units giving the greatest volumes at the proper temperature (at or slightly above room temperature) will prove to be economical.

The temperatures referred to are easily obtainable from the usual 50-125 lb. steam pressure available in almost all dye house plants. Since the unit heaters must be exposed to a low temperature and a fairly high steam pressure, their durability under such conditions warrants special consideration.

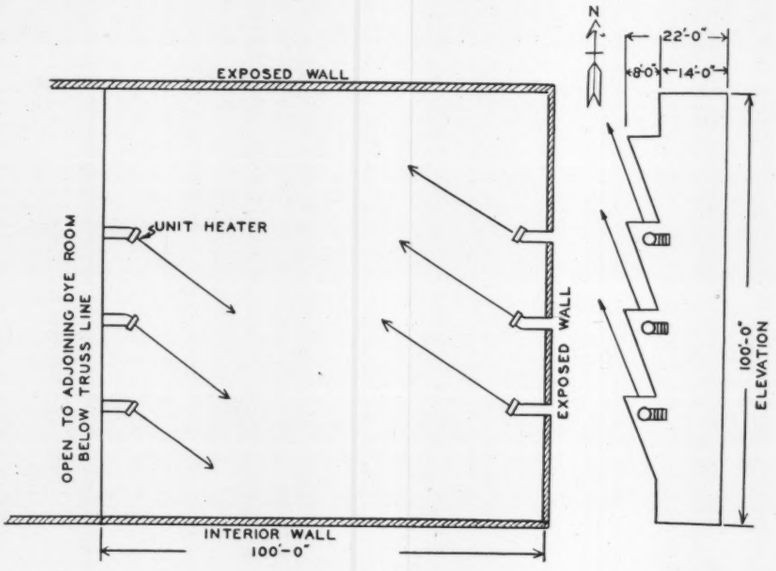


FIG. 1—FLOOR PLAN OF THE DYE ROOM AT THE WALDRICH BLEACHERY, DELAWANNA, N. J. TOTAL HEATING LOAD = 6 x 604,700 BTU ESTIMATED AT ZERO OUTSIDE TEMPERATURE = 3,628,200 BTU PER HOUR. STEAM PRESSURE, 80 LB.

They should be, of course, of such character as to resist freezing in case the steam supply fails and should also be immune to corrosive influences generally found in dye rooms.

A Typical Case

The simplicity with which problems of this nature can be properly solved, is, perhaps, best illustrated by giving pertinent data for several more or less typical installations.

For instance, at the Waldrich Bleachery, Delawanna, New Jersey, there was a room, 100′ × 100′, saw tooth construction, truss height 14′-0, and peak height 22′-0, as shown in Fig. 1. Two sides were exposed and one side was open to an adjoining dye room, but for purposes of estimating requirements, this was disregarded, since that room was handled separately.

The cubical contents were calculated at approximately 172,000 cu. ft. and because it was desired not to use exhaust fans, 15 air changes were decided upon. This meant that approximately 2,380,00 c.f.h. of outside air had to be introduced with a temperature rise from zero to approximately 75 F.

Selection of Units

Units raising 7258 c.f.m. from zero to 78 deg., operating under 80 lb. steam pressure, with face velocity of 1,264 ft. per minute, were used, being qualified in other ways to meet the needs of the occasion.

This velocity was of importance, because high velocity together with large volumes governed the uniformity of air distribution throughout the room, which was a vital factor in this case.

Multiplying by 60 to reduce to an hourly basis, we found that one unit would handle 435,480 c.f.h. and six units would handle 2,612,880 c.f.h. which was slightly more than required. (Five units would have probably proved sufficient, but six units insured *uniformity of distribution.*)

Location of Units

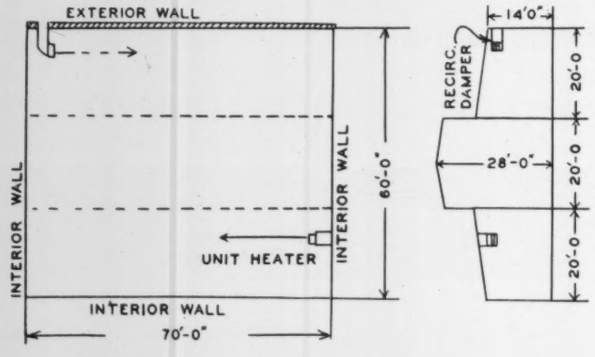
The triangular spaces forming the side walls of the saw tooth roof construction, seemed ideal for the location of the units and the units were accordingly hooked-up as indicated in the drawing.

The direction of air currents insures continuous circulation generally throughout the room which was, of course, essential, especially in view of the fact that no exhaust fans were used, the moisture-laden air being forced out of vents placed in the roof for that purpose.

The units were hung below the truss line, in order to prevent any interference with air movement and connected with ducts to the outside air, having a face area approximately the same as the face area of the unit, 30 in. × 30 in. or 900 square inches.

This was an important point, as the particular propeller type fans used, while very efficient for free air delivery, are not adapted for use when duct areas are sufficiently small or the duct is long enough to build up a considerable back pressure.

FIG. 2—FLOOR PLAN OF THE DYE ROOM AT THE GAEDE SILK CO., PATERSON, N. J. TOTAL HEATING LOAD = 2 x 589,200 BTU ESTIMATED AT ZERO OUTSIDE TEMPERATURE = 1,178,400 BTU. STEAM PRESSURE, 70 LB.



Another Installation

The installation at the Gaede Silk Dyeing Co., at Paterson, N. J., is also of interest. Here we had a room 70 ft. × 60 ft. of the monitor type, one side only exposed to outside air, adjoining rooms three sides, roof trusses 14 ft. 0 in. high with a monitor roof 28 ft. 0 in. high. The cubic contents were approximately 78,400 cu. ft.

Ten air changes were considered sufficient, which meant that 784,000 cu. ft. per hr. had to be handled.

For this purpose, two unit heaters were used, each raising 7,258 c.f.m. from zero to 76 deg., operating under 70 lb. steam pressure and driven by 1150 rpm motors.

The total air delivery amounted to $2 \times 7258 \times 60 =$ 770,960 c.f.h. which was considered close enough to requirements.

Recirculating Damper Installed

In this case, however, a recirculating damper was placed in the duct in order to give variable control of outside air and re-circulation of inside air, if such should prove advisable. This was, however, hardly ever used, and was not considered necessary when additional equipment was purchased for other rooms.

As shown in Fig. 2 one unit was placed in the duct leading to the outside wall and the other connected with the duct to the roof, both units being hung below roof trusses, with air current insuring a continuous circulation.

Because of the high monitor roof construction, a small amount of condensation developed in that section. This was due to the slowness of air movement over the cold roof construction and could have been prevented by the installation of exhaust fans opposite the unit heaters in the monitor or by increasing the volume of air introduced from the outside, either of which would have materially increased air movement, reducing the time of contact with the cold roof, which would have eliminated this objectionable feature.

The expense of additional equipment was not considered justifiable, in view of the small amount of trouble from the condensation that developed.

Measurement of Air Temperatures

By Malcolm Tomlinson

NO TEMPERATURE measurement is attended with so many difficulties as the accurate determination of air temperatures. There are many reasons for this. One is that unless air is kept in motion by mechanical means and unless the method of air movement is such that the circulation is uniform throughout the room in which the measurements are in progress, the air will pocket and considerable variation of temperature will be found. Another is that rays of heat (radiant heat), thrown off by various surfaces, raise the temperature of the sensitive element of the measuring device higher than the surrounding air temperature. A third difficulty is that hygroscopic insulations used to cover thermocouple wires often pick up moisture, which causes surface leakage. Other difficulties might be named but, as the three mentioned are those most often the source of trouble in work of an industrial nature, it seems better to discuss them than to discuss those of lesser importance.

How Radiant Heat Affects Temperature Measurement

Since the second difficulty named is the one which gives the most trouble, it will receive first attention here.

In recording air temperatures one deals with convected heat. This kind of heat is picked up by the air in passing over the surfaces of warm objects. This heated air travels, even under gravity conditions where mechanical methods of movement are not employed. It moves in a definite manner which depends upon such factors as the rate of air infiltration around doors and windows, the arrangement of objects in the room, the construction of the room itself and the location of the device which furnishes heat to the room. It, therefore, becomes imperative that no radiant heat rays fall upon the sensitive member of the measuring equipment.

Right here it may be well to point out that radiant heat is given off by hot or warm objects by reflection

and refraction, that it travels in straight lines and that such heat does not need to be "red hot" in order to be transmitted.

Size of Thermocouple Sensitive Member Affects Accuracy

Formerly, engineers who realized the seriousness of this problem, endeavored to solve it by the use of various types of shields. Modern research by instrument makers has shown another approach. The smaller the sensitive member the less it can be affected by rays of radiant heat simply because fewer rays of heat strike it. Therefore, where the thermocouple type of pyrometer is used, it is now considered good practice to use the smallest diameter for the two wires which form the couple that it is possible to employ. Therefore, whether the couple is made of copper-constantan, iron-constantan, platinum-iridium or some other combination, the former standard ⅛ inch diameter has been discarded in many installations in favor of smaller sizes of wire such as No. 24 American Wire Gage (0.020 inch dia.). For extreme accuracy, wires of much smaller diameter are recommended.

As the sensitive element of a thermocouple is at the welded joint or bead where the two metal wires are joined or fastened together, it is evident that the size of the bead depends on the diameter of the wire in the couple. Therefore the smaller the bead the lower will be the error due to radiant heat.

It is interesting to note that the radiant heat error in thermocouple measurements with ⅛ inch diameter wire is approximately 10 per cent of the temperature measured. In other words, under such conditions, a temperature indicated as 200 F might be, actually, somewhere between 190 and 210 F. As the diameter of the wire is reduced the error decreases.

One type of temperature measuring instrument consists of a volatile liquid, enclosed in a cylinder, which actuates a spring mechanism through a capillary tube. The sensitive element, in the cylinder, is located at the position where temperatures are desired. The minimum diameter of this element and the length of the cylinder afford much greater surfaces to receive radiant heat rays than are found in thermocouples.

Insulation of Thermocouple Wires

In measuring air temperatures accurately it has been shown that radiant heat rays are often a serious source of error. Most thermocouple wires are insulated. Where such insulation consists of a material with a smooth surface, such as porcelain, reflection of radiant heat rays which impinge from such surfaces on the bead of the couple will create an error of sufficient magnitude to affect accuracy. Therefore a type of construction must be used, for the end of the couple, which eliminates this danger. Such a type of construction is shown in Fig. 1.

Surface Leakage

Surface leakage of electric current due to moisture in hygroscopic insulations used on thermocouple wires is always a serious problem in electrical equipment which operates at low potentials and low currents. In thermocouple work, electrical leakage would create an error in the indicated or recorded reading. Where the relative humidity of the surrounding atmosphere is quite low (say under 10 per cent) this leakage is markedly reduced, but low humidities are the exception rather than the rule. It is best, where such hygroscopic insulation is more

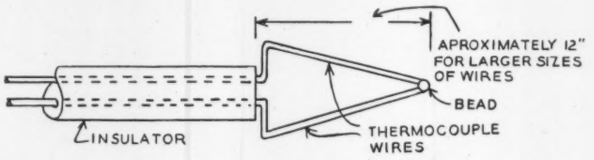


Fig. 1—End of a Thermocouple

practicable than porcelain, to provide a glazed surface for the insulation which will not crack when the insulation is twisted, with the thermocouple wires, into various shapes. The joker in this scheme is that there are almost no materials available for coating hygroscopic insulations which are flexible enough for the purpose and, at the same time, non-hygroscopic. Possibly one exception would be cellulose acetate.

Many people, in making servants of air, steam and water, try to compel them to do things which are contrary to all natural laws. This seems to be especially true in the use of air. Many large industrial plants have atmospheric dryers in which the air is short circuited and never has a chance to pick up moisture from the materials to be dried. The frictional losses in air ducts are often terrifically high, due to elbows which are too short or to obstructions which cause the air currents to buck themselves. It is not surprising, then, that many installations where mechanical air motion is used will not afford anything like uniform air temperatures. In gravity installations it is not hard to understand that considerable differences of temperature may exist.

There can be no doubt that the job of measuring air temperatures accurately, under many gravity and mechanical movement installations, is a problem which involves not only the proper measuring equipment but also the proper locations for the sensitive elements in the equipment. In the latter case no rules can be laid down. A temperature survey must be made, through observations in numerous locations, in order to determine the best positions for the sensitive elements in order that a true average temperature can be obtained.

Mercury thermometers, used for ordinary air temperature measurements, should be calibrated for total immersion and be readable to ¼ of a degree Fahrenheit.

Thermocouples of copper-constantan wire are used for from low temperatures up to 300 F. Iron-constantan couples give excellent accuracy from 300 to 900 F. At 900 F the platinum-iridium or precious metal couple becomes the most accurate and, from 1500 F to 2700 F is a satisfactory medium for accuracy in temperature measurement.

Determining Proper

Air Conditions for Libraries

To properly condition air for libraries it is necessary to study the characteristics of papers, leather and textiles. One source of information is the regain of moisture by these materials which is shown by the curves in the chart. Here sheepskin, silk, cotton, rag paper, newspaper and commercial ledger paper regain curves are compared.

The three papers, in the order noted, consist of 100 per cent rag stock, 100 per cent chemical wood stock and a combination of 75 per cent rag stock mixed with 25 per cent chemical wood stock. Other papers furnish regain curves somewhat similar to those shown so that they can be considered as characteristic or typical. All of these regain curves are based on room temperature slightly above 70 F.

Examination of these regain curves shows very clearly that the lowest rate of moisture regain for textiles and paper lies between 30 and 60 per cent relative humidity while that for sheepskin appears to lie between 50 and 70 per cent. It is in this region that these materials are least subject to change in characteristics of a physical nature. It is also evident that silk and sheepskin hold more moisture than paper or cotton throughout the range of relative humidities. This indicates that they are more seriously affected by low humidities. As far as leather is concerned, this is borne out by an examination of leather-bound books which have reposed for long periods in libraries which have no air conditioning. Most of these leathers have dried out to such an extent that their condition is serious while the paper leaves are in much better condition.

Investigations by the *Forest Products Laboratory* have demonstrated that changes in the dimensions of papers are practically zero at 60 per cent relative humidity and also that the folding endurance of paper is at a maximum at 65 per cent relative humidity. Wilson and Fuwa have also proved rather conclusively that the equilibrium water content of materials such as papers and textiles is constant at room temperatures between 72 and 77 F.

From this information, it is evident that the desirable conditioning for libraries would be a relative humidity of about 60 per cent with a temperature which did not vary much from the range of 72 to 77 F. If the reader will turn to page 220 of the March HEATING, PIPING AND AIR CONDITIONING he will find, in Figs. 1 to 3 inclusive, information which will enable him to establish the proper air motion at which human comfort can be secured with such humidity and temperature conditions.

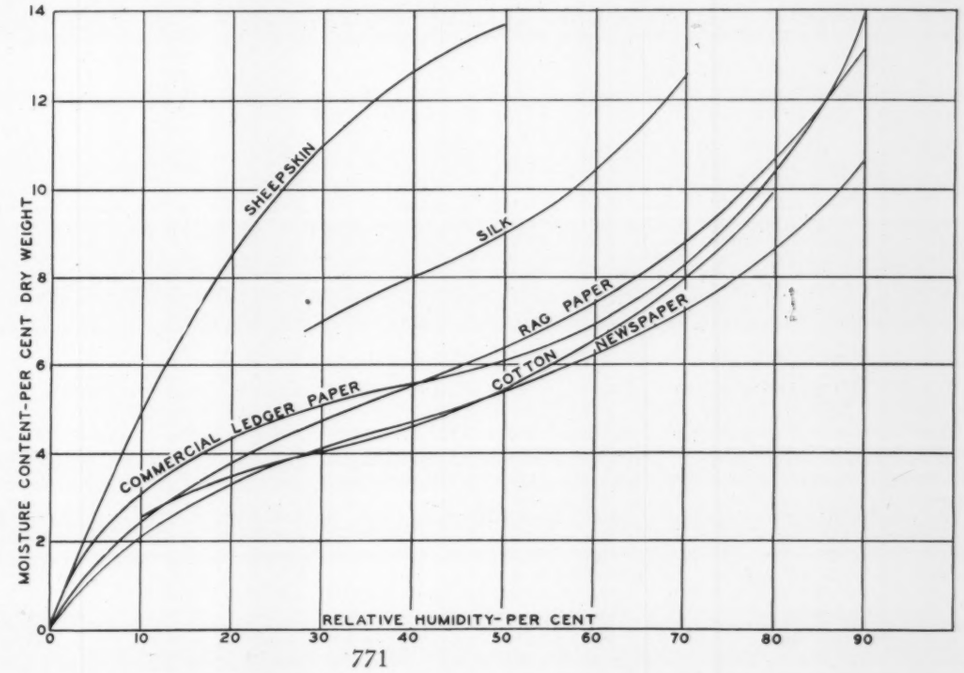
From data submitted by Yaglou and Drinker to the *A. S. H. V. E.* in January, 1929, the effective temperatures which give the greatest human comfort for human beings normally clad are 66 F in winter and 71 F in summer. The following table is based on the above data and shows the most suitable dry bulb temperatures for various rates of air motion which are normally used and for a relative humidity of 60 per cent:

WEATHER CONDITIONS SUITABLE FOR LIBRARIES AT 60 PER CENT RELATIVE HUMIDITY

F. P. M.	*100*	*200*	*300*
	Dry Bulb Temp. F		
Winter	71	73	75
Summer	77	79	80

The explanation of human comfort and the term effective temperature will be found in the article referred to in connection with the charts.

It must not be forgotten that, in addition to the preservation of manuscripts and books, air conditioning for libraries means comfort for the reader and the elimination of dust from books, shelves and the atmosphere.—*R. F. Morrison.*



MOISTURE REGAIN CURVES FOR SHEEPSKIN, SILK, COMMERCIAL LEDGER PAPER, RAG PAPER, COTTON, AND NEWSPRINT PAPER

"Open for Discussion"

A department in which we follow the custom of technical societies in allotting space in their programs for discussion of the papers presented

Woolen Mill Humidification

"Operating a Woolen Mill Humidifying System Correctly," *by James W. Cox, Jr.* Page 645, August, 1930, issue

THE article in the August issue by James W. Cox, Jr., should be of especial interest to every dealer in air conditioning equipment, as well as to those who are practicing its use. As a consultant textile expert he stands between the user of equipment on the one hand and the dealer who supplies it on the other. His experience and comments are therefore unprejudiced and entitled to especially serious consideration.

Humidifying Equipment Neglected

It is not too much to say that such neglect and mismanagement of air conditioning equipment as Mr. Cox has described are not only frequent but usual in almost any textile factory embarking for the first time upon the practice of humidification. Productive textile machinery nearly always receives whatever attention is needed to maintain its productivity; but humidifying equipment, although it contributes in a large measure to the maintenance of quality and quantity in the product, is usually subject to neglect until the operative personnel has been thoroughly educated to the importance of its proper maintenance.

It is for this reason that some dealers in air conditioning equipment maintain continuous inspection service in order to educate their customers to the necessity of the proper care and operation of the equipment in which their capital has been so freely invested and thus assist them to secure the manufacturing economies that will return their investment when the equipment is operated properly.

No other way seems open to dealers to protect their reputations and maintain their good-will, but the burden of expense entailed is by no means light and adds to the cost of humidifying equipment. Every dealer should welcome the activity of consulting textile experts who are detecting and exposing, as in this instance, the penalties and losses due to neglect of humidifying equipment, for they will thus assist their clients to secure the economies that are the reward of good management.

Mr. Cox makes a timely comment upon the practice of humidification in woolen and worsted manufacture in contra-distinction to cotton, which merits further discussion.

Advantages of Air Conditioning for Cotton Easily Apparent

In the cotton industry, humidification has become a universal practice. This is chiefly because the absorption and retention of hygroscopic moisture in cotton while being processed can be accomplished only by conditioning the factory atmosphere with respect to humidity. The manufacture of cotton products is, moreover, so seriously impaired as to both quality and rate of production by deficient or variable humidity that it may be fairly stated that a cotton mill could not be operated at a profit today without good humidification.

More Complex for Wool

The case for humidification in the woolen and worsted industries, on the other hand, is of quite a different nature. The advantages of air conditioning are no less real and profitable, but they are, unfortunately, less easy to explain and to visualize because they spring from more complex origins.

To illustrate: In the woolen industry, the raw material—as Mr. Cox points out—is charged heavily with emulsions before going through the mixing pickers. The material in process consequently carries—or ought to carry—not only a heavy content of hygroscopic moisture but also a certain additional content of free moisture.

That is to say, during picking and carding the material must be not only moist but actually wet in order to insure that physical condition of the fiber that is conducive to perfect processing. Reasonable stability in the *degree* of moisture and wetness at all stages of these important preliminary processes is essential to uniformly good processing.

Further, and even more important, there can be no effective control of weight per unit of length and in "count" of roping during carding, and yarn after spinning, unless the degree of wetness—that is, the percentage of moisture content—is held to reasonable stability at every stage of processing.

This leads inevitably to the conclusion that the application of emulsion to stock must be in a constant and proper predetermined ratio, and that atmospheric conditions must be so stabilized that there will be no such excessive or variable rate of evaporation as would deplete or vary the moisture content of the material to any important degree while processing continues.

A sensitive weighing device controls the rate of stock-feed to the cards, and thus regulates the weight per linear unit of roping, but this device cannot distinguish variations in moisture content of the material, nor compensate for them. It follows that material which is over-wet must necessarily result in roping which will contain too little actual fiber. Conversely stock which is over-dry means roping that is certain to contain too much actual fiber.

It is an inevitable consequence that so long as variations or deficiencies in atmospheric moisture are allowed to prevail in the carding department, they will be reflected in a highly variable roping because of variable evaporation.

The spinner is constrained to alter the "stretch" of

the mules in an effort to produce yarn of uniform weight and count from variable roping. Such measures, however, can only palliate without remedying the persistent variations in yarn count that are the constant bane of woolen and worsted manufacturers.

Losses Avoidable by Proper Air Conditioning

These ever-present sources of imperfection and financial loss are avoidable by the proper practice of air conditioning. It strikes at the root of the difficulty and prevents variation instead of trying to cure it after it has developed. It also secures all of the corollary advantages which Mr. Cox enumerates, and upon which I need touch only to say, that they are facts which have been repeatedly proved.

I have seen the weavers in a large cotton mill become so discouraged that they left work and the mill was obliged to shut down while the humidifying equipment was temporarily paralyzed by the breakdown of a pump. A similar result would be likely to follow in some departments of a worsted mill using the French process of carding and spinning. While such disastrous results might not occur in a woolen mill, or in a worsted mill using the Bradford process, the sudden cessation of proper humidification would produce results almost equally harmful, although processing could be continued after a fashion.

Waste Causes Large Financial Losses

Wool, as a material, and fabrics made from it, are so costly that the wastes which occur as the result of poor processing, and the financial losses which result from imperfections in the product or variations in its unit of weight, will be found to run into astoundingly high figures when they are studied and analyzed intelligently. It is because the causes of such losses are somewhat obscure and hard to visualize that their importance is less well understood than are the sources of loss which, in the case of the cotton industry, are more readily recognizable.

It is a real pleasure to endorse Mr. Cox's statements and to add emphasis to his comments upon the possibilities of air conditioning in the wool-using industries where its advantages are as yet too little understood.—*A. W. Thompson.*

City Smoke

"Smoke and Fumes —A Problem." Page 625, July, 1930, issue

YOUR editorial on smoke abatement in a recent issue is very much to the point. Smoke is no longer fashionable.

The day was when our leading citizens could "point with pride" to the smoking stacks of his town as evidence of community well-being. Today we realize its deterring effect on men and material, on the growth of children and the value of property. Smoke is more than a "nuisance"—it is a menace of no small proportions.

In my opinion the remedy is largely in public education as to proper methods of burning coal, campaigns of education rather than more laws.

The public obligation of our engineers in this respect is as clearly defined as that of the doctor or chemist. We, too, can contribute a distinct service to the advancement of community health.—*H. C. Murphy.*

Superheated Water

"Superheated Hot Water Circulation." Page 183, February, 1930, issue

AFTER reading the article on "Superheated Hot Water Circulation" in the February HEATING, PIPING AND AIR CONDITIONING, I have been trying to collect additional data and actual figures applying to the use of the injector mentioned therein, but as yet have not been fortunate enough to find any other treatise on this subject.

I presume that you are in possession of a great deal more information on this subject than you divulged in your article and would also be in a position to direct my search for information to sources in European literature which I have hitherto been unable to discover.

I am taking the liberty of stating and outlining my problem on a separate sheet, but I fear that it might be too much of an imposition to ask you to analyze it as to operativeness. Can you point out where I could find test data and a complete calculated example which would enable me to draw conclusions for application to the

The Problem

Total Btu of building = 343,000 Btu

Lb. water per hour and 30 F drop = $\frac{343{,}000}{30\text{ F}}$ = 11,433 lb. water

1/3 water supplied @ 225 F = $\frac{11{,}433}{3}$ = 3,811 lb.

3,811 lb. × heat of liquid (224) = 853,000 Btu

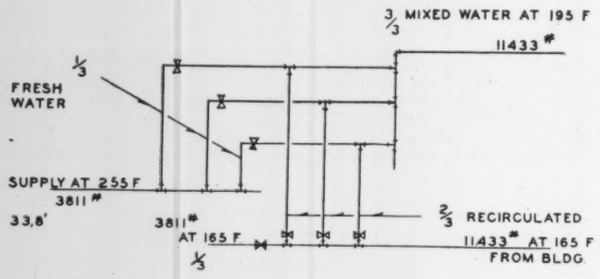
2/3 water recirculated @ 165 F = 7,622 lb.

7,622 lb. × heat of liquid (134) = 1,020,000 Btu

1,020,000 Btu + 853,000 Btu = 1,873,000 Btu

1,873,000 Btu ÷ 11,433 lb. = 164 Btu per lb.

From steam tables—164 Btu/lb.—195 F temp. of water.



above problem.—*Edward C. Caton,* Designing Engineer, The Austin Company.

The Author's Reply

Replying to your letter in regard to "Superheated Hot Water" I would advice that while in Russia I ran across this arrangement which was designed to be used in one of the heating plants installed for two 1929 apartment houses. There were two additional apartment houses which were to have been erected this year that were to have been hooked on to the same plant and in the same manner.

As explained to me in Russia the custom was to figure about one-third of the water coming over from the boiler plant at high temperature during extreme weather. The advantages claimed were reduction in size of feed and return mains, low heat loss from these mains owing to small size, low first cost of installation and a saving in power by pumping only one-third of the water against a slightly greater head resulting in reducing the size of the pump and motor as well as power consumed.

The mechanical department of "Mosstroy," one of the big construction organizations in Moscow, claimed to have used this arrangement on several jobs with entire satisfaction. As I was not there during the winter I had no opportunity to investigate the workings of this arrangement. Superficial examination rather inclines me toward the idea that, if the normal circulating head were figured on the pump plus the velocity head necessary to generate the *additional* velocity through the injector nozzles, this would be all the increase in pump head that would result. Of course the pump delivery would be reduced to 50 per cent or to 33 1/3 per cent according to the temperature used on the "superheated" water. In fact, the idea could be worked without the injector fitting at all, being entirely controlled by the valves in the small pipes and by the valves in the main lines to the boilers; because if a certain portion of very hot water were delivered into the end of the building supply main and a similar quantity of cooler water was withdrawn from the end of the building return main, it is only a matter of time until all the cooled water in the building has been replaced by hotter water from the boilers.

It is quite probable that the injector fittings serve the three purposes of accelerating circulation in the building lines, of causing enough return water to flow into the supply line to temper the superheated water and to provide for a thorough mixing of the supply and recirculated water so that all the water going into the building system is at the same temperature.—*Harold L. Alt.*

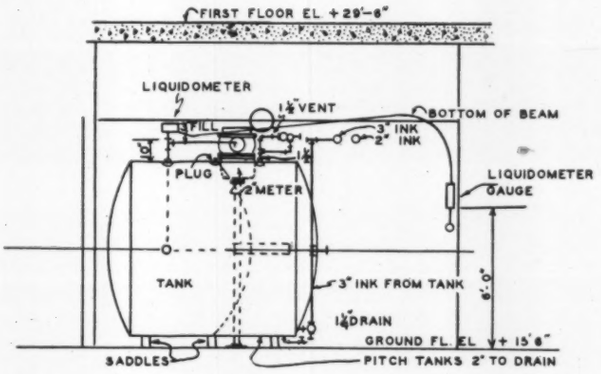
Reliable Ink Supply to Newspaper Presses Essential

THE average metropolitan newspaper today publishes about a half-million or more copies daily. Ink by the barrels is required, and obviously the old-fashioned methods of distributing ink to the presses are highly inadequate. Printer's devils in the modern newspaper office are about as quaint a spectacle as the lamp-lighter or the town crier. Presses can no longer be inked by hand; the operation is far too laborious and requires too much time. The ink must be fed by machinery that never fails, for the newspaper must be on the streets without a minute's delay.

A typical method of distributing ink—one which has proved satisfactory in every way—is to be found in the pressroom of the Chicago Daily News in its new skyscraper home on west Madison street. This plant has often been called the largest and the most modern in the world. The pressroom alone is three stories high and occupies about 20,000 square feet of floor space. It houses three banks of gigantic presses of fifteen units each, which require daily 1,800 hp. On an average day, eight barrels of ink are consumed by these presses.

Ink Comes in Tank Wagons

The ink is brought to the building directly from a manufacturing plant in tank wagons. A hose is coupled to the wagon at the building and is attached to the ink supply in a separate tank room located on the east side of the ground floor underneath the driveway platform and the presses. The ink is conveyed to this room by a pump attached to the tank wagon, through three three-inch fill lines. One of these lines is connected to each of the tanks. Before the ink reaches the tanks it passes through a set of three-inch strainers, so that all foreign particles may be removed which might clog the meters. These gages are operated by a float system and in-



Elevation of Ink Tank Showing the Necessary Connections to Each Tank, as Installed in the Chicago Daily News Building

dicate accurately the quantity of ink in each of the tanks.

From the tanks the ink is forced to the pressroom through a three inch pipe. This pipe, which has screwed fittings, tapers to a two-inch diameter about half way down its length, then to an inch and a half, and finally to one inch. In the pressroom the pipe is connected with smaller ink lines running parallel to the rows of presses. Brass risers run from these small pipes to the ink boxes or fountains at each press unit.

Forcing Ink to Presses

Perhaps the most interesting feature of this system is the method of forcing the ink from the tank supplies on the ground floor up through the pipes and ultimately to the presses. There are three such tanks, each having an approximate capacity of 2,500 gallons. While one of the tanks is being filled, the other two may be in use, since the fill lines are separate. At the top of each tank is a 1½-in. vent and an air line running to an air compressor. At the bottom is a by-pass and a pump connected to the tank by a suction line.

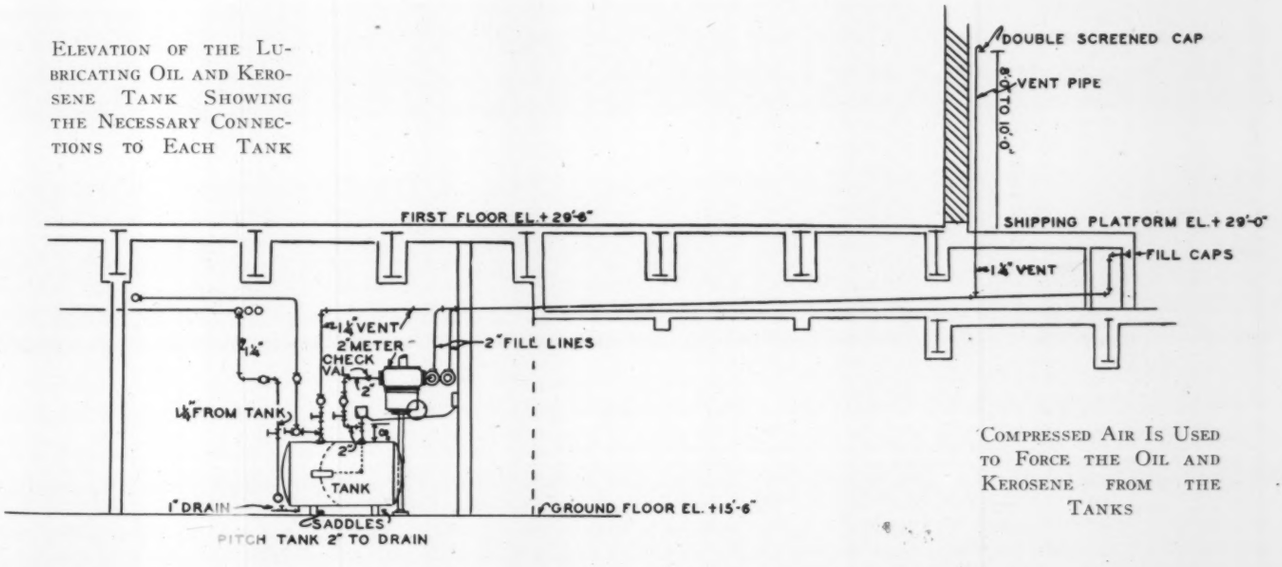
Either of two methods is used in forcing the ink from these tanks into the pipes that lead to the pressroom. When the pump is in operation, the by-pass valve is closed, as are the valves in the air pressure and the ink fill line. The vent at the top of the tank is then left open. When the pump is not in use, the ink is forced into the main pipe by air pressure. The ink supply line in either case must be closed. When the compressor is in operation, the vent is closed also, and the pump is cut out. The ink is then forced out of the tank and through the by-pass into the main pipe. The air pressure used varies according to the height to which the ink is to be raised. Ordinarily, about thirty or thirty-five pounds are required, but if the ink is to be forced to the third floor of the pressroom, more than forty pounds of pressure is used.

Supplying Kerosene and Oil

The same general arrangement and operation is installed in the building to deliver kerosene and lubricating oil. The kerosene and oil is carried from the outside of the building through two-inch fill lines equipped with strainers and meters. Four 250 gallon tanks, two for each system, are located in a room adjoining the ink storage room to the south. Compressed air only is used to force the oil and kerosene from the tanks, although provision has been made for future installation of pumps should they prove necessary. The oil and kerosene then pass through a series of piping which supplies risers at the columns in the pressroom and the stereotype room. Near the floor line special fill faucets are provided in the risers, where workmen can tap off any amount of oil and kerosene required for lubricating or cleaning purposes. The faucets are set within a few feet of the presses and other machinery, so that a minimum of time is necessary for workmen to keep the machinery in perfect running condition.

Benzine is drawn from a 300 gallon storage tank encased in concrete and buried under the basement floor of the building. The fill box for the benzine is at the sidewalk above.—*J. V. Dodge.*

Elevation of the Lubricating Oil and Kerosene Tank Showing the Necessary Connections to Each Tank



Compressed Air Is Used to Force the Oil and Kerosene from the Tanks

Piping Engineers Need Information *on* Hydraulic Equipment

By F. G. Schranz* and W. L. DeLaney*

SINCE MAN first discovered and applied the principle of the simple lever, whereby with a relatively small force he was able to overcome a large resistance, he has searched for means of increasing his sphere of action. The use of water for irrigation and similar purposes was known to remotest antiquity, but it remained for the last century to see the discovery and development of the application of water under pressure to doing useful work. Since the first successful application by Bramah in 1795, its sphere of usefulness has constantly increased until at the present time there are few, if any, industries in which hydraulic equipment does not play an important part.

The maintenance engineer in a large Chicago industrial plant says, "It is absolutely essential for a good hydraulic pipe man to understand the hydraulic machinery for which he is installing the piping, or which comes under his supervision.

"It is necessary for the hydraulic pipe engineer to know all the parts of the equipment. The capacities of the rams, the speed of the presses, and all other points must be considered carefully when determining the size of the pipe lines.

"Naturally, there are different opinions as to how the piping should be installed, but all the methods of piping are based on the design and use of the press."

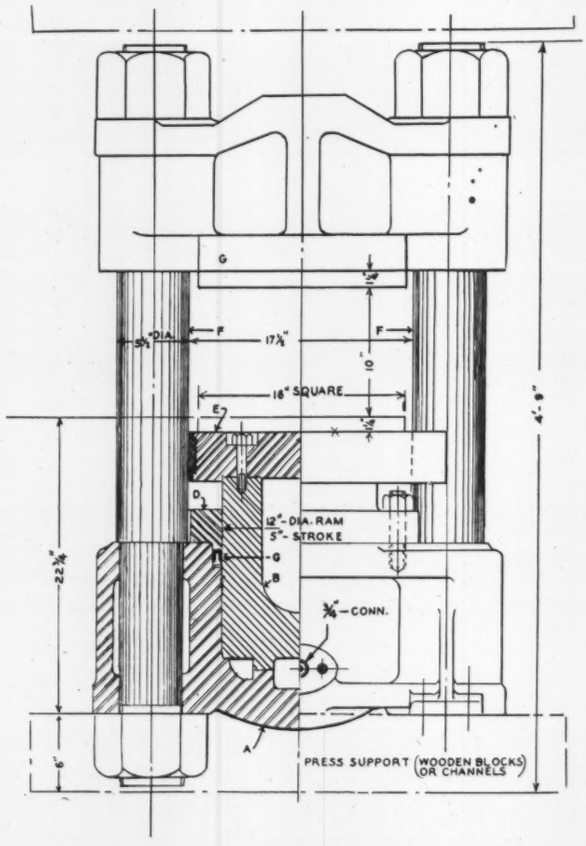
Not only are present systems being maintained, but additions are constantly being made and as new industries develop, new applications and installations follow. The facts are attested by sales of extra strong and double extra strong pipe, aside from its other markets where such weights of pipe are used for high pressure, as in the oil refining industry and refrigeration, or where there is excessive corrosion as is met in the chemical industry.

Hydraulic power in its simplest form consists of a pressure vessel (usually termed the cylinder) fitted with a plunger or ram, having a packing box to retain the fluid, together with a means for supplying water or oil under pressure. An hydraulic-type elevator, where the load is carried directly on a platform on the end of the ram is one of the simplest and most frequent applications.

The Hydraulic Press

An hydraulic press consists of a pressure vessel with ram and fluid supply for same at pressure and, in addition, a structure to permit applying the power developed. This structure in its simplest form consists of a head tied rigidly to the cylinder by two or more tension members (columns) and a platen mounted on the end of the ram and moving with it. The moving platen may or may not be guided against the columns.

Presses may be classified in a diversity of ways, but whatever system may be chosen, it is customary to speak of the capacity as if so many tons. This represents the power of the press to overcome resistance in doing useful work and is determined by the area of ram and the fluid pressure which is applied. The accompanying table shows the capacities of various rams (diameters are given for simplicity) at various pressures. They are based on the following formula:



BELOW—A FOUR-COLUMN HYDRAULIC PRESS. *A*, PRESSURE VESSEL; *B*, PLUNGER OR RAM; *C*, PACKING; *D*, PACKING GLAND; *E*, MOVING PLATEN; *F*, TENSION MEMBERS ON COLUMNS; *G*, TOP HEAD

* Southwark Foundry and Machine Co.

CAPACITIES OF HYDRAULIC RAMS IN TONS FOR VARIOUS DIAMETERS AND PRESSURES

	PRESSURE IN POUNDS PER SQUARE INCH																					Gals. per 1" Stroke	Gals. per Ft. Stroke	Area
DIA.	300	500	600	750	1000	1200	1500	1800	2000	2250	2500	2750	3000	3250	3500	4000	4500	5000	5500	6000	6500	7000			
2	.47	.8	.94	1.2	1.6	1.88	2.4	2.82	3.2	3.6	4	4.3	4.7	5.1	5.5	6.4	7.2	8	8.6	9.4	10	11	.013	.163	3.14
3	1.05	1.7	2.10	2.5	3.4	4.20	5.3	6.3	7.0	7.8	8.5	9.6	10.5	11.4	12	13.6	15.9	17	19.3	21	22	24.5	.030	.364	7.06
4	1.88	3.1	3.76	4.7	6.2	7.52	9.4	11.28	12.4	14	15.5	17	18.8	20.5	22	24.8	28	31	34.5	37.6	41	44	.054	.650	12.58
5	2.94	4.9	5.88	7.4	9.8	11.76	14.8	17.6	19.6	22	24.5	27	29.5	31.5	34	39	44	49	54	59	63	69	.087	1.02	19.64
6	4.25	7.1	8.50	10.9	14.2	17	21.8	25.5	28	32	35.5	39	42.5	45	49	56	65	71	78	85	90	99	.122	1.47	28.27
7	5.73	9.5	11.46	14.3	19	22.92	28.5	34.38	38	43	47.5	52	57	62	67	76	85	95	105	114	124	134	.167	2.00	38.48
8	7.50	12.5	15	18.7	25	30	37.5	45	50	56	62.5	69	75	85	92	100	112	125	138	150	170	184	.227	2.73	50.26
9	9.55	16.	19.10	23.7	32	38	47.5	57	64	71	80	87	95	104	112	128	142	160	174	190	208	223	.277	3.30	63.62
10	11.8	19.6	23.6	29.5	39	47	59	71	78	88	98	108	118	128	137	156	172	195	217	236	256	275	.341	4.10	78.54
11	14.2	23.6	28.4	35.5	47	56	71	84	94	106	118	130	142	154	166	188	213	235	260	284	308	332	.410	4.92	95.00
12	17.	28.	34.	42.	56	68	84	102	112	126	140	155	169	183	197	224	252	280	310	338	366	395	.487	5.85	113
13	19.9	33	39.8	49.5	66	79	99	119	132	148	165	186	198	214	230	264	297	330	362	396	428	460	.571	6.85	132
14	23	38	46.	57.5	76	92	115	138	152	172	190	210	230	250	270	304	345	380	420	460	500	540	.667	8.00	154
15	25.4	44	50.8	66	88	101	132	153	176	199	220	242	264	285	307	352	396	440	484	528	570	615	.775	9.30	176
16	30.	50	60	75	100	120	150	180	200	225	250	275	300	325	350	400	450	500	550	600	650	700	.867	10.40	201
17	34.	56	68	85	112	136	170	204	224	255	280	313	340	370	397	448	510	560	625	680	740	795	.983	11.80	227
18	38.	63	76	95	126	152	190	228	252	285	315	349	380	414	445	504	570	630	698	760	828	890	1.10	13.20	254
19	42.5	71	85	106	142	170	212	255	284	318	355	388	425	460	495	568	636	710	775	850	920	990	1.22	14.64	283
20	47	78	94	117.5	156	188	235	282	312	352	390	430	470	510	550	624	705	780	860	940	1020	1100	1.36	16.32	314
20½	49.5	82	99	123.5	164	198	247	297	328	370	410	455	495	535	575	656	741	820	910	990	1070	1150	1.43	17.16	330
21	52	86	104	130	172	208	260	312	344	390	430	475	520	560	605	688	780	860	950	1040	1120	1210	1.50	18.00	346
21½	54.5	91	109	136	182	218	272	327	364	408	455	500	545	590	635	728	816	910	1000	1090	1180	1270	1.57	18.84	363
22	57	95	114	143	190	228	285	342	380	429	475	521	570	620	665	760	855	950	1042	1140	1240	1330	1.64	19.68	380
22½	59.5	99	119	148	198	238	297	357	386	444	495	545	595	645	695	792	891	990	1090	1190	1290	1390	1.72	20.64	397
23	62	103	124	155	206	248	310	372	412	465	515	570	621	675	729	824	930	1030	1140	1242	1350	1458	1.80	21.60	415
23½	65	108	130	162	216	260	325	390	432	486	540	597	650	705	760	864	975	1080	1195	1300	1410	1520	1.88	22.56	434
24	68	113	136	170	226	272	340	408	452	510	565	621	680	739	795	904	1020	1130	1242	1360	1478	1590	1.95	23.52	452
24½	70.6	118	141	177	236	282	355	423	472	531	590	650	709	765	825	944	1065	1180	1300	1418	1530	1650	2.04	24.48	477
25	73.6	121	147	183	242	294	367	441	484	541	605	670	735	800	860	968	1101	1210	1340	1470	1600	1720	2.12	25.44	490
25½	77	127	154	192	254	308	387	462	508	576	635	700	765	830	895	1016	1176	1270	1400	1530	1660	1790	2.20	26.40	510
26	79.6	132	159	198	264	318	397	477	528	594	660	730	795	865	930	1056	1191	1320	1460	1590	1730	1860	2.30	27.60	531
26½	83	138	166	207	276	332	415	498	552	621	690	760	830	895	965	1104	1245	1380	1520	1660	1790	1930	2.40	28.80	551
27	86	143	172	215	286	344	430	516	572	645	715	785	860	930	1000	1144	1290	1430	1570	1720	1860	2000	2.48	29.76	572
27½	89	148	178	222	296	356	445	534	592	660	740	820	890	965	1040	1184	1335	1480	1640	1780	1930	2080	2.56	30.72	594
28	92	153	184	230	306	368	460	552	612	690	765	845	920	1000	1075	1224	1380	1530	1690	1840	2000	2150	2.66	31.92	615
28½	95.5	159	191	238	318	382	477	573	636	714	795	875	955	1040	1120	1272	1431	1590	1750	1910	2080	2240	2.75	33.00	638
29	99	165	198	247	330	396	495	594	660	741	825	910	990	1075	1155	1320	1485	1650	1820	1980	2150	2310	2.85	34.20	660
29½	103	170	206	255	340	412	510	618	680	765	850	935	1020	1115	1200	1360	1530	1700	1870	2042	2230	2400	2.96	35.52	683
30	106	176	212	265	352	424	530	636	704	795	880	970	1060	1150	1240	1408	1590	1760	1940	2120	2300	2480	3.06	36.72	707
31	113	188	226	287	376	452	565	678	752	851	940	1035	1130	1220	1315	1504	1695	1880	2070	2260	2440	2630	3.26	39.12	754
32	120	202	240	302	404	480	605	720	808	906	1010	1108	1210	1300	1400	1616	1815	2020	2216	2420	2600	2800	3.47	41.64	804
33	128	213	256	320	426	512	640	768	852	960	1065	1175	1280	1380	1490	1704	1920	2130	2350	2560	2760	2880	3.70	44.40	855
34	136	226	272	340	452	544	680	816	904	1026	1130	1250	1360	1470	1585	1808	2040	2260	2500	2770	2940	3170	3.91	46.92	908
35	145	240	290	360	480	580	720	870	960	1080	1200	1320	1449	1560	1680	1920	2160	2400	2640	2880	3120	3360	4.16	49.96	962
36	152	253	304	380	506	608	760	912	1020	1140	1275	1388	1520	1650	1780	2024	2280	2530	2776	3040	3300	3560	4.38	52.50	1017
37	161	270	322	402	540	644	805	966	1080	1206	1350	1475	1610	1750	1885	2160	2415	2700	2950	3220	3500	3770	4.63	55.50	1075
38	170	283	340	425	566	680	850	1020	1140	1275	1415	1555	1700	1850	1985	2264	2550	2830	3110	3400	3700	3970	4.91	58.92	1134
39	179	298	358	447	596	716	895	1074	1192	1341	1490	1640	1790	1940	2090	2384	2685	2980	3280	3580	3880	4180	5.15	61.80	1194
40	188	313	376	470	626	752	940	1128	1252	1410	1565	1725	1880	2030	2195	2504	2820	3130	3450	3760	4060	4390	5.41	64.92	1256
41	198	330	396	495	660	792	990	1188	1320	1485	1650	1815	1980	2140	2300	2640	2970	3300	3630	3960	4280	4600	5.70	68.40	1320
42	208	346	416	520	692	832	1040	1248	1384	1560	1730	1900	2080	2250	2425	2768	3120	3460	3800	4160	4500	4850	6.00	72.00	1385
43	218	363	436	545	726	872	1090	1308	1452	1635	1815	2000	2180	2350	2535	2904	3270	3630	4000	4360	4720	5070	6.28	75.36	1452
44	226	380	452	570	760	904	1140	1356	1520	1710	1900	2087	2280	2470	2650	3040	3420	3800	4175	4560	4940	5300	6.57	78.84	1520
45	238	396	476	595	792	952	1190	1428	1584	1785	1980	2180	2380	2580	2775	3168	3470	3960	4360	4760	5160	5550	6.87	82.44	1590
46	250	416	500	625	832	1000	1250	1500	1664	1875	2080	2290	2500	2700	2900	3328	3750	4160	4580	5000	5400	5800	7.38	88.56	1662
47	260	433	520	650	866	1040	1308	1560	1732	1950	2165	2390	2600	2820	3025	3464	3900	4330	4780	5200	5640	6050	7.50	90.00	1735
48	271	451	542	677	902	1084	1355	1626	1804	2031	2255	2485	2710	2940	3150	3608	4065	4510	4980	5420	5880	6300	7.80	93.60	1809
49	283	471	576	706	942	1152	1412	1728	1884	2118	2355	2590	2825	3060	3300	3768	4236	4710	5180	5650	6120	6600	8.30	99.60	1885
50	295	490	590	735	980	1180	1470	1770	1960	2205	2450	2695	2940	3190	3425	3920	4310	4900	5390	5880	6380	6850	8.50	102.00	1963
51	307	511	614	767	1022	1228	1535	1842	2044	2301	2555	2810	3070	3310	3550	4088	4605	5110	5620	6140	6620	7100	8.75	105	2043
52	319	531	638	797	1062	1276	1595	1914	2124	2391	2655	2925	3190	3490	3705	4248	4785	5310	5850	6380	6900	7410	9.30	112	2120
53	330	551	660	827	1102	1320	1655	1980	2204	2481	2755	3035	3310	3590	3875	4408	4965	5510	6070	6620	7180	7750	9.55	115	2206
54	343	573	686	860	1146	1372	1720	2058	2292	2580	2865	3100	3440	3720	4000	4584	5160	5730	6300	6880	7440	8000	9.80	118	2290
55	358	593	716	890	1196	1432	1780	2148	2392	2670	2965	3265	3560	3860	4185	4784	5340	5980	6530	7120	7720	8310	10.21	123	2376
56	370	616	740	925	1232	1460	1850	2220	2464	2775	3080	3395	3700	4000	4310	4928	5550	6160	6790	7400	8000	8620	10.61	127	2453
57	382	637	764	960	1274	1528	1912	2292	2548	2880	3185	3500	3825	4140	4475	5096	5736	6370	7000	7850	8300	8950	11.00	132	2551
58	390	661	792	992	1322	1584	1985	2376	2644	2976	3305	3640	3990	4280	4625	5288	5955	6610	7280	7940	8600	9250	11.40	137	2642
59	410	683	820	1025	1366	1640	2050	2460	2732	3075	3415	3750	4100	4400	4775	5464	6150	6830	7500	8250	8880	9550	11.80	142	2734
60	425	708	850	1062	1416	1700	2125	2550	2832	3186	3540	3900	4250	4600	4900	5664	6375	7080	7800	8500	9200	9800	12.32	147	2827
PRES.	300	500	600	750	1000	1200	1500	1800	2000	2250	2500	2750	3000	3250	3500	4000	4500	5000	5500	6000	6500	7000			
Lbs. per Sq. In.	.175	.291	.350	.438	.583	.700	.875	1.05	1.167	1.32	1.46	1.60	1.75	1.90	2.05	2.35	2.63	2.92	3.21	3.50	3.80	4.08			

$$T = \frac{\pi D^2}{4} \times W.P. \div 2{,}000$$

T = Capacity in tons

D = Ram diameter in inches

$W.P.$ = Fluid pressure in lb. per sq. in.

π = 3.1416.

Applications

Industry has evolved many distinctive types of presses for specific application and a great many others for special processes. This type of equipment is especially applicable to processes requiring full power over considerable periods of time, or distance in contrast to the mechanical types, such as power or toggle presses, which are used where maximum power is developed through a limited range. In applying the hydraulic press to any given problem it is necessary to know the tonnage required. In simple cases, such as bending a steel bar, the power may be readily determined, whereas for a process such as corrugating plate where both stretching and compression may take place, the final power would probably be the result of experimentation.

According to a manufacturer of hydraulic equipment "a number of customers who have bought hydraulic equipment from us have had trouble in getting contractors to install the piping correctly, as a majority of piping contractors have little idea of the functioning of the hydraulic press. We have had a number of instances when it was necessary to send a service man to correct costly mistakes."

It is the purpose of this article, and future articles on the same subject, to supply this information.

In order to give an idea of the extent to which hydraulic presses are used, a list of industries who are using hydraulic press equipment is given below. Later articles will review some of the special and common applications.

Abbatoirs and Stockyards; Abrasive Wheel Manufacturers; Aluminum Manufacturers; Animal Oil Manufacturers; Asbestos Manufacturers; Asbestos Shingle Manufacturers; Automobile Plants; Artificial Silk Manufacturers; Bag Manufacturers; Bakelite Product Manufacturers; Brass and Copper Works; Battery Box and Jar Manufacturers; Boiler Shops; Bone Industry; Bridge Builders; Briquetting Plants; Car Shops; Carbon Manufacturers; Casein Products Manufacturers; Celluloid Product Manufacturers; Ceramics; Chemical Works; Chocolate Manufacturers; Composition Board Manufacturers; Copper Works; Cork Industry; Cotton Baling Plants; Die Sinking Shops; Embossing Plants; Engraving Plants; Electrotyping Plants; Extruded Metal Shapes; Felt Manufacturers; Fertilizer Factories; Fibre and Fibre Goods; Food Products; Forge Shops; Frog and Switch Shops; Garages with Tire Presses; Gear Manufacturers; Grinding Wheel Plants; Hard Rubber Manufacturers; Hose Manufacturers (Rubber); Jewelry Manufacturers; Lampblack Manufacturers; Leather Manufacturers; Linoleum Plants; Machine Tools; Mines and Collieries; Oil Pipe Line Pumping Plants and Refineries; Paper Mills; Pressed Steel Plants; Powder Mills; Poultry Feed Manufacturers; Pipe Works; Printing Works; Pulp Mills; Range Boilers; Rayon Works; Railroad Shops; Salt Industry; Soap Industry; Textile Manufacturers.

INTERVIEWS OF INTEREST

An Index Chart for Steam Plant Operation

Charles J. Lehn, chief engineer for the Marshall Field Estate, and responsible for management routines in its various properties, has devised a steam plant operation chart so simple to use, so readily adaptable to changing situations, and so uniformly useful that only a heating engineer could surmise how much patient work and painstaking investigation have entered into its construction.

All building engineers who serve the Marshall Field Estate utilize the chart. Many building managers are inclined to adopt it as a standard. It condenses in a single graphic statement the essential facts that relate to legitimate steam consumption and reduces the aggregate power load to terms of hourly distribution. It is, in effect, a simple index of exactly how the plant must be operated in order to maintain economically the required temperatures, and it protects against unsafe and costly fluctuations in use of steam. It favors cost accounting. It eliminates the guess-work of night operating engineers as to when steam should be taken off and when the morning engineer should put it on.

Mr. Lehn has no notions about how a building shall be run without minute study of its operating conditions. "Go out and get the evidence," he says. "It is on the basis of actual operating conditions that the individual operation routines are built up."

With respect to steam consumption the "evidence" is detailed and highly specific. The annual power load is determined by experience and by meter readings. The distribution of this load is then worked out by months, by weeks, and, finally, hour by hour. Regulation of steam pressures is made on basis of the hourly load within the building in conjunction with readings taken of outside temperatures.

Power consumption within the building is rhythmic; the minimum load is noted at the same hour every day; the work done is cumulative and consistent. The peak power load, too, is at the same hour every day and the peak load for any single hour is never higher than 9 per cent of the total twenty-four hour steam consumption.

This relates to office and apartment buildings and it applies only to radiation steam and not to heating air for ventilation. "Any factor that raises the basic hour-by-hour steam load as much as 1 per cent above this normal for a whole day's operation becomes a serious economic matter in building management," states Mr. Lehn; "hence the value of employing a graphic chart and daily index to reveal such changes promptly. A visual index is much more impressive than any study that can be quickly made of tabulated costs."

In the construction of his graphic chart (Fig. 1), Mr.

Lehn employs a quad-rule plot. Vertical figures at the left carry a temperature reading for each line, beginning with zero degrees at the top and increasing 3 degrees for the reading of each line until 70 degrees is recorded on the bottom line.

The three horizontal rows of figures at the top of the chart indicate: Row 1, the number of hours of service required by the radiation to maintain the required temperature.

Row 2 expresses the service hours in terms of percentage of use that will be required of radiation.

Row 3 is the hourly scale which, in the final analysis, shows the precise hours when steam is turned on and off.

Line *AB* on the chart is the summation curve of actual operating conditions within the building. The curve will vary somewhat from building to building. Its use is to set up the hours that are necessary for the radiation in the building to be in service. It is used thus: At zero degrees, the use of radiation would be 100 per cent, or 24 hours of service. When outside temperature is 12 F, the hours of operation would be between 80 per cent and 85 per cent of service, or between 19.2 and 20.4 hours. The hours to turn steam on and shut it off are found by the engineer in the following manner:

Follow the horizontal line marked 12 deg. to point where it crosses line *CD*. From this point of interception trace the vertical line upward to the horizontal figures at the top. Reading in Row 1 shows the necessary hours of daily operation; Row 2 shows percentage of radiation use; Row 3 indicates time steam may be turned off. To find hour when steam must be turned on for efficient heating under this condition, follow the horizontal line through to the interception point in line *EF*, and then trace the vertical line upward to the proper reading. At 24 F outside temperature, about 70 per cent of the radiation would be required to heat the building, or 16.2 hours of steam out of the twenty-four. Heat would be turned on about 4 A. M. and off at 8 P. M.

Mr. Lehn considers zero degrees as the point of 100 per cent utilization instead of –10 deg. as is usually accepted. In his operation chart for practical purposes line *CDEF* is flattened off at *DE* at 12 deg. on the assumption that full radiation will be called for from this point. The scale of *CDEF* as indicated in the chart shown will not fit every building, but very small adjustments in the

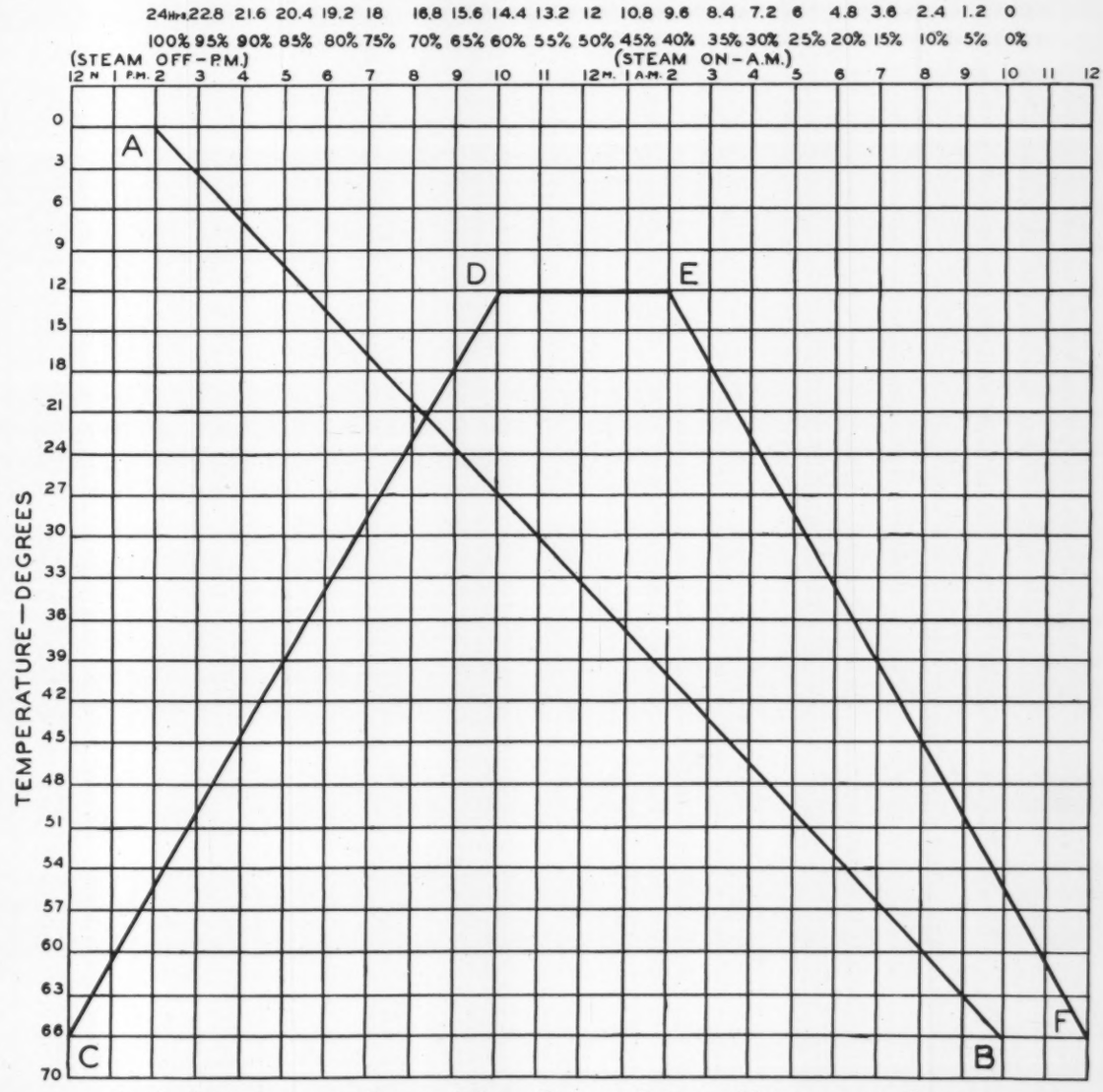


FIG. 1—CHART DEVISED BY CHARLES J. LEHN PROMOTES PRECISION MANAGEMENT IN BUILDINGS

angles of inclination of *CD* and *EF* will enable suitable readings for other buildings.

With the consistent use of this chart, either with or without automatic temperature control, building temperature will not go below 60 deg. no matter what the outside temperatures may be.

"The matter of heating costs is highly variable," states Mr. Lehn, "but for the most part fluctuating costs in building operation are much more likely to relate to personnel than to strictly mechanical factors.

"There are many variables. In buildings *A* and *B*, two structures under present observation of the type described, we find that heating steam for *A* per year per cubic foot of space was 4.9 lb., for *B* it was 6.1 lb. Heating steam per square foot of rentable area was 103.7 for *A*; and 109.1 for *B*.

"But this relationship was reversed when comparisons were made between heating steam supply per square foot of radiation. This figure was 487.7 pounds for *A*, and 355.3 pounds for *B*. The changed relationship was due to differences between the glass areas and the exposed wall spaces, the cubical content being the factor for this."

Mr. Lehn's system of collecting evidence has resulted in the accumulation of much highly pertinent general information with respect to legitimate building costs for water and electricity as well as for steam. We have noted that in any single hour any legitimate use of steam would not exceed 9 per cent of the total twenty-four hour load. For tenants' use, one gallon of hot water and one and one-half gallons of cold water per person will be required per day. This means that 16 cubic feet of total water will be required per year per square foot of rentable space. Any amounts of water used over and above these figures other than the janitor and engineering requirements may be charged to waste.

Electricity consumption for commercial tenants' use will run 2¼ kw to 4½ kw per year per square foot of rentable space. Total electricity consumption, including general lighting, power, elevator service, and the like, will be from 5 to 7 kw per year per square foot of rentable space.

Mr. Lehn's graphic records are interesting. On many more points than these we have described his series of charts set up an actual picture month by month of the operating conditions he observes routinely. Each chart always visualizes two years of record for every comparison he desires to make on matters of building management. There is individuality in his methods, and he earns his illuminating perspectives by careful work.

Indiana University Chemistry Building

In connection with his general engineering services on the new chemistry building at the University of Indiana at Bloomington, Charles R. Ammerman, consulting engineer, faced a many sided problem in developing a system of hygienic air control.

The long main building in front is four stories in height. At either end there is a three story wing, each devoted to class room uses. The building is constructed of Bedford limestone. It has been erected at a cost of $650,000.

The structure will house all the chemistry classes on the campus; all types of chemical operations had to be provided for. Fifteen separate systems of piping have been installed to take care of its several services.

Mr. Ammerman states that the fresh air supply for this building is supplied by two separate systems. The two three-story wings devoted to class room use receive air by means of window unit ventilators of the recirculating type. In the main building with its laboratories, air is supplied through a central system with filters of the panel type.

Gravity exhaust is utilized for class rooms. "There is nothing in the Indiana code," states Mr. Ammerman, "to hamper effective engineering design, and exhausts can be kept small enough not to interfere with building up the desirable positive pressures within the occupied rooms."

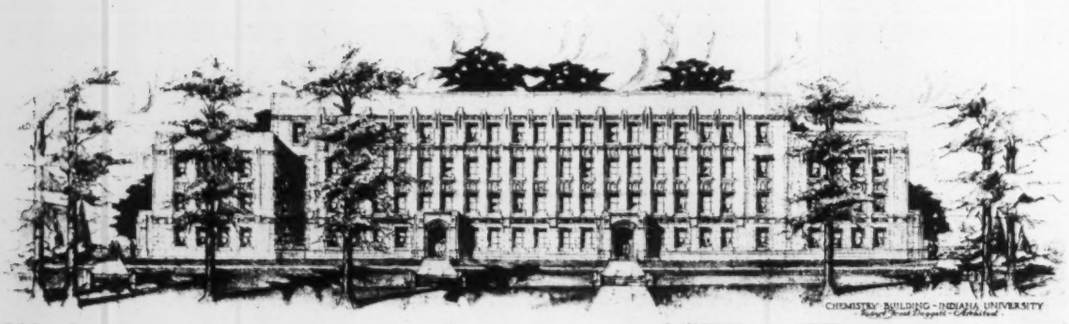
Fan exhaust is utilized for the main building. Four separate exhaust systems are arranged: (1) Special exhaust from down-draft hoods of laboratory tables only; (2) exhausts from attic space into which the exhaust air from class rooms, with no chemical fumes, is carried; (3) exhaust system for toilets and other service rooms; and (4) exhausts from large wall type fume hoods.

The air capacity from the main fume hood is large enough to take care of room ventilation and so to dilute all exhaust air that none of the general exhausts requires fans of acid resisting construction.

Acid resisting tile is used in all inaccessible ducts, and also in the vent in the chemical laboratory flue.

Lead coated, copper bearing steel is used as a base in duct work and after construction all duct work was sprayed inside and out with acid resisting lacquer.—*S. P. M.*

New Chemistry Building at the University of Indiana at Bloomington. The Architect Is Robert Frost Daggett



Heating·Piping
and Air Conditioning

1895

American Society of Heating and Ventilating Engineers

JOURNAL SECTION

SEPTEMBER 1930

Material in the Section prepared by Publication Committee: G. L. Larson, *Chairman*; H. H. Angus, and W. A. Rowe : A. V. Hutchinson, Manager of Publications

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Air Conditioning for Summer and Winter Comfort

By W. H. Carrier, Newark, N. J.
MEMBER

The author has discussed the progress in the art of heating and ventilation and foresees in the near future the application of scientific air conditioning to homes bringing to the average house holders at the snap of a switch an indoor winter climate that will rival the pleasant climate of Florida and in summer the invigorating, embracing air of the seashore or mountains.

In discussing this paper one of Detroit's leading nose and throat specialists, Dr. Simpson, said that it was heartening to find the engineering profession meeting the complex problems of modern living conditions.

Dr. Simpson pointed out that it was most interesting to try to bring about a better living condition for the providence of good health, and it has been a very disappointing thing also. It has been noticeable to him that the families working for the production of better air can do much through properly heating the air up to what we know as reasonable temperatures under which mankind can live and work, and then adding to that air a sufficient amount of moisture so that they will be less subject to these infections.

Dr. Simpson also said, "It is a great field for you to contribute to the health of the people living in a climate where we must heat our homes artifically in the winter, also where we need very much to cool them in the summer. It is a most useful field and a chance to be of service to your fellowmen. You will do the communities in the northern and temperate zones a good that the whole medical profession alone cannot do."

HEATING and ventilation is an art and, like other mechanical arts, began with crude but practical methods of accomplishment in the earliest history of the human race. It has become progressively more complex and scientific as demands have become more exacting, and economic needs have warranted more highly refined developments.

The general art of heating and ventilation, from which sprung the highly specialized art of air conditioning, has two distinct fields of application. The earliest, and perhaps the most important, is heating and ventilation for human health and comfort, while the more recent, and in some respects the more exacting field, is in its application to industrial arts and manufacturing processes.

This paper is confined to the first of these two distinct fields, that of air conditioning for human health, comfort, and efficiency and more particularly to present advanced practice in this field although, with a discussion of the trend of development in this art and the possibilities of the future.

The Human Power Plant

In considering the requirements of the human body, the most useful method of approach for the engineer is to compare it with a power plant, which in fact it is.

It is the function of the human body to take fuel into the alimentary canal, which may be considered the pulverizer and magazine for the fuel, and to burn this fuel after being distributed through the arteries to the tissues of the body, which are the furnace. The air for combustion, and the products of combustion, are both handled by a combination of forced and induced draft through the lungs and respiratory system. The energy produced by the combustion of the fuel with the oxygen of the air is transferred into mechanical energy and heat with a considerable measure of efficiency. This transformation, as well as the attained efficiency, is in strict accordance with the laws of thermodynamics which indicates that only a portion of the available heat energy can be transformed into work, while the greater portion must always be expelled from the system in the form of heat at a lower potential; that is, the entropy of the system always increases.

Just as in a power plant, it is necessary not only to supply oxygen to the body and to remove the products of combustion, carbon dioxide and water vapor, but it also is necessary to remove the large proportion of unavailable heat which is present in every thermodynamic power cycle.

In the steam power plant, to which the human body may be compared, there must be a condenser in which the heat is removed by water or other heat conveying medium. In the case of the body, this heat conveyor is the blood. This cooling liquid must be pumped through the condenser or heat absorber. In the body the heart is the pump. When the heat in the power plant has been transferred from the condenser to the water, means must be provided for again cooling this water before recirculation. In the power plant operating on a closed cycle this water is cooled by radiation, conduction, and evaporation in a cooling tower through which the cooling water must be circulated in the closed cycle. Similarly in the human body, there must be a cooling tower—the skin (and to a much smaller extent, the lungs). It is a function of heating and ventilating

Presented at April Meeting of the Michigan Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

engineers to provide for the removal of the waste products of the human power plant, the waste heat and the products of combustion.

The Human Cooling Tower

In the early days of the art, the principal function of ventilation was considered to be the removal of the waste products of combustion, and little thought was given to ventilation as a means of removing the waste heat.

According to present available information there are never insufficient quantities of oxygen present or an excess of waste products in the air of a ventilated room when the requirements of air for removing the heat from the body are met; that is, the requirements for operating the cooling tower of a human power plant are far in excess of those for supplying the oxygen chemically required for combustion just as it is in the mechanical power plant.

In ventilation, or air conditioning, for human comfort the prime consideration is, therefore, the removal of heat from the human body by means of radiation, conduction and evaporation, just as it is the prime requisite in the power plant. The efficiency of the human power plant is perhaps higher than that of other animals. There is perhaps a smaller percentage of waste heat as compared with useful energy. It is said that James Weston, the walker, could, over a period of days, out-distance and leave a horse in a condition of exhaustion.

Warm blooded animals are usually provided with some form of insulation against too great a rate of heat flow from the skin (the human cooling tower). This insulation in some animals is variable, the winter coat being heavier than the summer coat; but man through adaptation and evolution is more sensitive to external temperature changes and has produced for himself a variable artificial insulation by means of which, within quite wide limits, he can control at will the rate of waste heat conduction from the body. On the other hand, the animal has a great power to compensate for increased heat losses by a higher rate of metabolism, that is a higher rate of combustion. Man does this to some extent, but has lost much of this power of accommodation through evolution, although it is possible he never possessed it; having originated in tropical or semi-tropical climates where the problem was one not of conserving the heat of the body but of its disposal. Perhaps of all animals, with the possible exception of the horse, man is best able to stand high temperatures. The perspiratory system of the ape is very feeble compared with that of man. A monkey exposed to the extreme heat of the sun for any length of time would die while man would scarcely be affected.

Man, in changing his natural environment from warmer to colder regions, has learned two things. *First,* to produce for himself, means of varying his insulating covering at will, by means of clothing of different weights and thicknesses thus adapting himself to the severe temperature changes of the temperate climates. *Second,* he, of all the animals, has devised means of producing locally at least some simulation of the climate he requires. This has proceeded step by step. He learned the use of fire, probably at first for warmth, and later for cooking. Then he learned the value of fire in an enclosure as compared to fire in the open so that he could conserve the heat and regulate the warmth to a desired degree by the quantity of fuel consumed. At this stage he probably had no problem of ventilation because the open fire in an enclosure, such as an Indian tepee, required plenty of ventilation to keep the air free of smoke from the burning wood. Neither was he interested in cooling for comfort, except as he might plunge into the lakes or streams, or seek the refuge of a cool cave, or expose himself to the breezes with his clothing removed. Civilized man today, however, has so surrounded himself with artificial conditions and has set up such standards of living, that he requires, or at least finds desirable, much more than did the primitive man, and this is as true in the field of ventilation and heating for comfort as it is in the matter of clothing, foods and entertainment.

Regulating Body Heat Losses

Considering that clothing serves as an insulator, preventing excessive heat losses and regulating them within measure, why heat or cool? Why not regulate entirely by the clothing? This is done in China, which has variations in climate comparable to our own. But efficiency is not obtained by working in overcoats, nor do the proprieties of modern civilization permit people to go naked in their offices during a hot day, although they do remove their coats—and the ladies have reduced their clothing to a short 2 lb, including their shoes. At night when requirements are not so severe, enough heat insulation can be piled on in the form of blankets and quilts to keep warm in an un-heated room or sleeping porch in the middle of winter, thus maintaining a normal heat outflow from the body. In summer a sheet may be sufficient, although even then discomfort may be experienced on certain hot oppressive nights.

It is true that it is possible for the human body, to a large extent, to control its own heat output and to keep it within the limits of normal metabolism[1] even without the artificial variations in clothing or regulation of room temperature. This is done by means of automatic regulation of the perspiration poured out by the sweat glands. When the skin temperature tends to rise, these sweat glands open up and greatly increase the rate of evaporation[2] and, therefore, the removal of heat. The system cannot alter the rate of radiation and conduction, however, to any great degree, except by unusual changes of skin temperature. Controlling the amount of heat removed through perspiration is like the operation of the winter front of an automobile to maintain constant radiator temperatures. An extreme demand upon the human regulating system, however, produces discomfort. One is not comfortable when the extreme contraction of the pores produces goose-flesh on the body and the teeth chatter, nor is one comfortable when it is necessary to mop the brow continually, or when the clothing sticks, indicating that the skin—the cooling tower of the system—is being taxed to the utmost.

Between these two extreme conditions of discomfort, there are definite conditions of optimum comfort[3]

[1] Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities, by McConnell, Yaglou and Fulton, JOURNAL, A. S. H. V. E., January, 1925.

[2] The Theory of Atmospheric Evaporation, by W. H. Carrier, *Journal Industrial and Engrg. Chemistry,* Vol. 13, No. 5, May, 1921.

[3] The Summer Comfort Zone: Climate and Clothing, by Yaglou and Drinker, JOURNAL, A. S. H. V. E., January, 1929.

which are fairly constant for the average individual, but which vary somewhat between winter and summer climatic conditions owing to change of clothing and what may be termed acclimatization.

Present State of Ventilation Research

In the laboratories of the U. S. Bureau of Mines at Pittsburgh, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS maintains a laboratory in which various researches relating to the fundamentals of the heating and ventilating art are conducted. One of the most interesting series of researches which have received world wide recognition has related to heat losses from the human body under various atmospheric conditions and upon the proportion of heat that is lost by evaporation and radiation and conduction and how these factors vary with the change of temperature. In this connection also the relations of temperature, moisture and air motion to the sensation of comfort have been discovered and tabulated. A measure has been found in terms of which such conditions may be expressed. For example, it was found that a temperature of 65 F with still and saturated air is equally comfortable to a temperature of 76 F in air almost free from water vapor. Both conditions are said to have the same effective temperature and, for convenience, that the corresponding temperature of saturated air, in this instance 65 F, is used as the temperature of reference and called the *effective temperature*. Similarly, certain conditions of air motion can be compared in comfort so far as the feeling or sensation of temperature is concerned with still air saturated. Thus, for any condition of temperature, humidity and air motion there has been determined the corresponding effective temperature. This is entirely an experimental relationship tried on numerous individuals with very concordant results; it is nothing that can be determined directly by the laws of physics, or by the application of mathematics. However, the curves of effect can be graphically drawn in terms of wet-and dry-bulb temperature and air motion and these curves can be approximately expressed by empirical equations.

It has been found that the average human being, at rest, has a heat output of approximately 400 Btu per hour, which is the approximate equivalent of a 120 w electric light, and remains remarkably constant through all normal ranges of temperature and moisture variation and changes of clothing. The constancy of this heat output is determined by the relatively constant metabolism of the body. The control of heat removal equal to such metabolism is maintained with constant body temperature by means of the automatic variation in skin evaporation. Another remarkable fact observed is that the skin evaporation is not dependent upon the humidity of the surrounding air, as was previously supposed.

A certain amount of heat is transmitted from the body through the clothing by conduction, convection and radiation as sensible heat. The quantity of heat thus dispersed is determined entirely by two factors (1) the difference in temperature between the human body and the surrounding air and surrounding walls and (2) the clothing, which modifies in a measure the rate of such heat removed.

Since the heat given off by the human body is substantially constant except under extreme conditions, it is evident that the balance of the heat not taken care of by radiation and conduction through the clothing must be taken care of by evaporation through perspiration, and this is exactly what has been found to occur by these researches at the A. S. H. V. E. Laboratory. Therefore, the evaporation from the human body, assuming constant metabolism, depends entirely on radiation and convection to the surroundings, and this in turn is affected by the temperature of the surroundings and the clothing. At a temperature of 70 F a normal man at rest, normally clothed, will transmit about 76 per cent of his bodily heat as sensible heat by radiation and convection, while the balance of approximately 24 per cent will be dissipated through evaporation of perspiration and from the lungs.[4] This proportion is apparently in no way effected by the relative humidity of the surrounding air. Similarly at 86 F, about 38 per cent will be sensible heat, while 62 per cent will be removed by perspiration, and at a dry-bulb temperature of about 100 F all the heat in the human body must necessarily be removed by perspiration. In all such changes, no important change takes place in the amount of heat produced by the body and removed from it nor in the body temperature. While such changes do not affect the total cooling effect on the body, yet the accommodation of the body to these conditions gives a sense of discomfort due to the very slight changes of skin temperature.

For every individual normally clothed there is a certain optimum effective temperature at which he experiences the greatest comfort. This effective temperature for greatest comfort, which may be termed the *optimum effective temperature*, depends, of course, upon the clothing and it also varies with different individuals with normal clothing. The optimum effective temperature also differs between summer and winter. It has been found by researches at Harvard,[5] which were coordinated with the researches of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at the U. S. Bureau of Mines, that an optimum comfort line giving a maximum comfort of 97 per cent of individuals tested was 66 F for winter and 71 F for summer. Thus, the physiological requirements for ventilation, with respect to the removal of heat and moisture from the body and maintaining a maximum comfort, are quite thoroughly established, but there are many factors still to be determined in reference to the requirements of ventilation.

Synthetic Air Chart

A number of years ago, Dr. E. Vernon Hill, Chicago, developed what he termed a *synthetic air chart* in which the qualities of ventilation were given definite values and by which a composite ventilation value could be accorded by determining the departure from certain assumed standards. This method of rating ventilation was most constructive but it did show that definite scientific knowledge of the proper standards was lacking in practically every item. At that time no information concerning effective temperatures was available, nor were the lines of optimum comfort known, although Dr. Hill had made some investigations in this direction. It

[4] *A Study of Body Radiation*, by L. B. Aldrich (Publication No. 2080) Smithsonian Miscellaneous Collections, Vol. 81, No. 6.

[5] *The Summer Comfort Zone: Climate and Clothing*, by C. P. Yaglou and Philip Drinker. (A. S. H. V. E. Journal, Vol. 35, Jan., 1929.)

was for this reason that the studies of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on the relation of temperature, humidity and air motion to human comfort were undertaken.

Today, however, there are many other things remaining to be determined. The effect of extremes of atmospheric moisture on health are not known, nor are the optimum conditions of atmospheric moisture for health known. It is not known whether the variability of temperature promotes health. The percentage of fresh air to be introduced as an optimum from the standpoint of human well-being and consistent with economy has not been determined. Recirculation is being practiced with air conditioning but it is not known definitely how much to recirculate safely. The effect of dust in ventilated rooms is not known, either on normal human beings or those having respiratory diseases. Many suspect there is a quality of air which is neither determined by its oxygen content nor by any other chemical characteristic which vitally effects the well being of man. There are many indications that such an unknown vital property exists. This might be termed the *vitamin* of air. Many believe that this is associated with the ionization of air, an electrical property which is indicated by the effect on an electroscope and is said to be due to the partial separation of positive and negative ions of which the atoms are composed. It is most certain that such separation greatly increases its chemical activity. It is known definitely that outdoor air has a different ionization factor than indoor air and the presence of people in a room most markedly reduces the normal ionization, but whether it has any physiological effect is not known at all. Ventilation cannot be discussed scientifically until these most important fundamentals are known. Steps are now under way for studies along these lines, that is, of the effect of various atmospheric properties upon human well being. Researches such as these are primarily the concern of the physiologists and the medical profession. Some work of this nature is already in progress at Johns Hopkins and Harvard Universities. The engineers, for their part, have developed the methods and the equipment for creating and maintaining a combination of air conditions to make these investigations possible.

Status of Air Conditioning as Applied to Human Comfort

The growth of air conditioning was brought about largely by the requirements of industry. The effect of atmospheric moisture on materials, such as textiles and tobacco, has long been known to be favorable to increased production and improvement in the quality of the product. Many other products besides textiles and tobacco, require relatively high moisture conditions for the best production and quality of product. On the other hand, there are many products, such as confections, matches, and gelatine products, in which both high moisture and high temperatures are distinctly a detriment and natural conditions during summer months were frequently such as to make impossible continuous satisfactory operation of the plants without cooling and dehumidification.

But it was not until the art of air conditioning had thus been largely perfected to industrial uses that its value as a factor in human comfort and efficiency was recognized or applied. It may be said that the knowledge of the underlying scientific principles and the development of the technic as to the methods of application of air conditioning for industry paved the way for the wider application of air conditioning for human comfort. Also the researches at the A. S. H. V. E. Laboratory during the last eight years have definitely determined physiological effects of all ranges of atmospheric environment and more specifically have determined the definite requirements for optimum human comfort. Both these developments have thus made possible the intelligent and scientific application of air conditioning to promote human comfort and efficiency.

The major applications of air conditioning, as might be expected, have been in places where people are most thickly congregated, and where ventilation requirements thus become most exacting, as in theatres, hotels and department stores, and, in some instances, office buildings. In auditoriums and department stores it is necessary to contend primarily with the removal of heat from the bodies of the people to produce conditions conducive to comfort; while under conditions of less intensive occupancy, such as in office buildings, it is necessary to contend primarily with offsetting external weather conditions, conduction of heat through walls and, most important, the intense heating effect of direct sun-light through the windows and upon walls and roofs of the buildings. In all such air conditioning installations, three distinct factors must be considered, *First,* conditioning the air to the proper dew-point and temperature for ventilation requirements; *second,* the calculation of the factors requiring heat removal, such as heat from the bodies of the people, radiation and sun-light, and *third,* to offset these heat gains, there must be corresponding heat balance in cooling effect provided by the calculated quantity of air of predetermined condition necessary. A failure to consider the factors involved in any one of these three requirements means a failure in the air conditioning system as applied. Thus, air conditioning for human comfort during the summer time is a considerably more exacting science than merely ventilation with heating for winter.

Economic Aspects of Air Conditioning for Human Comfort

Theatres and department stores adopt air conditioning for business reasons. They have found it pays dividends. The first theatre that was equipped in New York City paid for the installation in increased box office receipts the first season.

Department stores have found they could crowd their stores on the hottest day in summer when neighboring stores were almost completely deserted. Air conditioning was a better drawing card in summer than advertising special sacrifice sales.

Air conditioning will become more prevalent in banking and private office buildings when it is found, as the author believes it will be, that the dividends paid by the greater efficiency of employes pays well for the cost of installation and operation. A citation of some random costs may be interesting as showing the economic feasibility of such applications.

Restaurant or Cafeteria, 1c per patron served.

Large offices, drafting rooms, etc., 1c per man hour.
Small offices, drafting rooms, etc., 3c per man hour.
Theatres, 3 2/3c per seat per day or 2c per patron.

Physiological Benefits of Air Conditioning

An air conditioning system for the summer makes possible also air conditioning during the winter and provides for comfort and healthfulness far superior to the ordinary conditions secured by steam radiation with gravity window ventilation, or even with the ordinary type of mechanical ventilation. With the normal type of heating system, in winter the humidities average altogether too low and temperatures have to be correspondingly raised too high for comfort. Automatic control of the dry-bulb temperatures, while possible, is not satisfactory for the reason that the effective temperature, which is the temperature that should be controlled, depends on moisture as much as on heat. Excessive dryness of the air in winter not only produces a parched condition of the nose and throat, and uncomfortable dryness of the skin, but also is a most fruitful cause of the production of dust. One can even smell dust when entering a dry, heated room. Such dust, while perhaps not harmful in itself, may produce an added irritation of the nose and throat and other parts of the respiratory system. This dust can be controlled to a large degree with humidification together with air purification where a portion of the air is recirculated.

The Future of Air Conditioning in the Home

The author predicts at this time that the future of air conditioning lies not in its application to theatres, department stores, and office buildings, but to the home. Here is where a great part of a person's time is spent and our children and wives spend most of their time. Here is where the most can be done to conserve the health and comfort of the people as a whole. The floors on which the children play should not be cold or draughty. Extremes of temperature and excessive variability should be prevented. Extreme dryness in winter is to be avoided. Relief from the torrid days of summer is desirable when it can be afforded. The home can be made more comfortable in summer than the finest summer resort, and with far less annual expense.

In the author's opinion, the ordinary steam or hot air heating system today is somewhat crude, as it is designed primarily for the lowest first cost rather than for either comfort or the best economy.

In the study of the progress of mankind, as exemplified in any of the mechanical arts, it is found that progress is made along one line of thought until nearly the utmost that such a line of endeavor can afford is realized, then it becomes necessary to make a sudden jump over to a new track, and abandon the old method for a new method of an entirely different character. The initial efforts in this new line are almost invariably crude, but they in turn soon become perfected and when their perfection is reached, a third similar jump is made, and so on through the cycle of progress. To mention a few instances of such typical developments, witness the progress in the burning pine knot and afterward through candle and lamp to the gas flame, and then the sudden jump to an incandescent light with its electrically-heated filament. The latter bears no resemblance to the early and gradually perfected method except that it radiates light. Following the same line of thought, it is to be expected that some day the present incandescent light will be superseded by other and better methods of illumination. Witness the transition in power transmission from the wheel, belt and pulley to electric transmission; from the steam power plant to the internal combustion engine; and so on along the line.

Viewing the art of heating homes as it is today, the author cannot but feel that the zenith of present methods has been approached. A point has been reached where its defects are being recognized, where the demand for human comfort and even luxuries outweigh considerations of cost. It is for this reason that the author predicts a rather abrupt change within the next 50 years in the method of obtaining bodily comfort in homes. Inevitably it will be some form of air conditioning equipment which will provide equitable heat in the winter with the proper degree of moisture and thorough cleanness of the air in the rooms as well as reasonable ventilation. It should be possible readily to cool homes in summer, to reduce the enervating humidity on sultry days, and to sleep soundly in cool bedrooms at night.

It is quite possible that gas will be the coming fuel because of its cleanliness, ease of regulation, and its real economy of raw material and transportation. It may be that electricity in some locations will also play its part, especially in connection with the application of the reversed refrigeration cycle.

In any event, it seems inevitable that some day the average householder will be able to press a button in winter and produce a climate in his home rivalling the pleasant days of Florida, and press another button in summer to bring the invigorating and bracing air of the mountains and seashore to his abode.

The Hot Weather Specialist

Nowadays everyone seems to be going into this specializing business. As long as we see this specializing trend we're going to get prepared to become specialists ourselves. The subject of just what to specialize in had us momentarily stumped. But finally we narrowed all possibilities down to one. We are hot weather specialists. A hot weather specialist is a person who can tell you how to keep cool during the hottest, hot spell. First of all you will have to buy an oblong water trough. For the best results get one about two feet long by one and a half feet wide by about eight inches deep. Place it under your desk, fill it with ice cold water from the office drinking fountain, take off your shoes and socks, roll up your trousers and place your feet right in it. If you wish, you may buy a few healthy gold fish to place in the trough in order to give it a realistic touch. The second thing to do is to buy one of those long palm fans and bribe the office boy to wave yours.

An ingenious phone attachment concludes our researches thus far. As you lift the receiver to talk, this spring attachment releases a thin, cool stream of ice cold water which sprays your face as you talk.

Should you have any of your own personal cooling systems that you would like scientifically investigated, send them in. We have plenty of time to do more research work. There is a crying need for better personal cooling systems and we are going to do our share.—*Elbe News.*

Application of Heating and Ventilating Research

By L. A. Harding[1], Buffalo, N. Y.
MEMBER

THE PROGRESS of any art, measured by the improvements made in the processes involved, may be said to be entirely regulated by the amount of well directed original research which is expended on the art. This is a fact, which only in comparatively recent years, has been fully accepted by the industries.

Engineers are happy to acknowledge the debt they owe to the work of natural philosophers and physicists, dating from Newton's time, which has made possible the practice of engineering in its many and varied branches. As new facts that find practical application in the general scheme of our existence are discovered, there arises the apparent necessity for men to study and practice new branches of engineering.

The rapidity with which the formation and growth of new engineering and technical societies has taken place during the past few years leads to either one of two conclusions, that this apparent need could be either satisfactorily met by membership in one of the existing major engineering societies, corresponding to the degrees conferred by our technical institutions, or that the present subdivision and degrees do not adequately serve the needs of this age of specialization in the application of physical laws.

The invention of the stove by the versatile Franklin in 1774 is generally accepted as the starting point in heating economy, convenience and comfort. The progress of this art has been practically continuous since his time and has closely followed the demands made by the invention of a Chicago architect, of the multi-story curtain-wall type building, the modern theatre and the many and varied demands from the industries. The prominent role which purely scientific and engineering research has played in removing the art of heating and ventilation from uncertain engineering endeavor is frequently overlooked.

Heat Transmission

The pioneer research of the French physicist Peclet, made nearly a century ago, covering the heat conductivity and emissivity of various materials, was perhaps the first contribution of real scientific value to this art. Heat transmission investigations and tests by many physicists and engineers have been more or less continuous since his time and there are now available conductivity values for practically all known building materials and heat insulations. Computed values of compound walls were for some time considered by this industry as somewhat academic. The Nichols heat meter developed in the laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been applied to a sufficient variety of compound walls of existing structures to definitely establish that it is a safe procedure to employ laboratory tests on materials in the calculation of the heat transmission of such walls.

The effect of wind on the combined coefficients of radiation and convection, termed surface coefficients of roofs and walls, and the consequent effect on the heat transmission of these areas, was investigated for a variety of wall surfaces in 1914 and 1915 by Harding, Willard and Lichty at the University of Illinois. The surface coefficients determined during that period have been widely adopted in this and foreign countries.

Practically no research covering this item has been conducted since that time until last year when the A. S. H. V. E. Laboratory undertook the work, under the direction of F. C. Houghten, of redetermining these coefficients, employing a somewhat different method of procedure. The correlation between laboratory velocity measurement and wind velocities in a free unobstructed space, as reported by the Weather Bureau, is now in the process of solution.

Research has furnished the engineer with a more accurate method for estimating the heat required to compensate for air infiltration based on tests of various kinds of sash with and without weather-stripping. The amount of infiltration through plain and plastered walls has been definitely established.

New Standards

In the past any heating plant that would maintain a breathing line temperature of 70 F, 5 ft above the floor line during coldest weather, was considered satisfactory. The occupants, however, are frequently not agreeable to this idea. They occupy a zone at an average height of 2 ft 6 in. above the floor line. It is often an observed fact that, although an air temperature of 70 F in the 2 ft 6 in. zone is maintained, the occupants are not in that state of well being or quiet enjoyment called comfort.

By *comfortable* is meant that a fairly exact heat balance is being maintained between the heat liberated within the body, by chemical reactions, and the heat leaving the body by radiation, convection and other ways. Any other condition tends to produce a change in the body temperature and the sensation of warmth or of coldness is experienced.

Power plant engineers have in recent years become quite *heat balance* minded. Heating and ventilating engineers have always been so afflicted, but were frequently unconscious of the fact. It is apparent that the functions of clothing and a heating system are identical, that of playing the principal role in the regulation of the amount

[1]President, AMERICAN SOCIETY HEATING AND VENTILATING ENGINEERS. Address presented before the Annual Meeting of the *National District Heating Association* at St. Louis, Mo., June 3, 1930.

of heat dissipated by the body. Either will serve the purpose, but with considerable less annoyance in cold weather when properly combined. Approximately 30 per cent of the heat dissipated by the body is lost by convection to the surrounding air, approximately 43 per cent by radiation, and 27 per cent by exhalation and other means when an air temperature of 70 F and 25 per cent relative humidity is maintained, with the surrounding wall surface temperature at approximately 70 F.

It has been a recognized fact for some time that various combinations of air temperature, air movement and relative humidity will produce equal degrees of comfort with the enclosure surface temperature kept relatively constant at approximately 65 F or above.

One of the greatest scientific contributions to the art of heating, in recent years, is the *comfort chart* established by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and which has satisfactorily solved one part of the problem.

A considerable percentage of body heat loss, however, is accounted for by radiation, which is dependent upon the surface temperature of the clothing and the surrounding wall temperature. The recent research of Professors Willard and Kratz, of the University of Illinois, on wall surface temperatures observed in building construction has served to focus the attention of engineers on this neglected, but important, part of the problem dealing with comfort.

It is somewhat complicated due to the effect of solar radiation and wind movement, particularly on walls having southern exposure. The insulation of wall surfaces, primarily to reduce the heat loss of the structure, also serves the equally important function of increasing the inside surface temperatures and therefore to retard the body radiation loss.

Further research will undoubtedly develop standard wall transmission units, which for various localities may be relied upon to give satisfaction in this respect. Recently the speaker listened to a lecturer who referred to an observed fact that research workers in the near vicinity of 6 meter radio wave generating apparatus experienced a considerable increase in blood temperature. He recommended a study of this phenomena to the heating industry as a possible substitute for present practice. The absorption of certain beverages is responsible, along with other frequently curious symptoms, for the same effect, but either method apparently involves a mistaken conception of the function of a heating system.

Sun Effect on Buildings

The absorption of solar radiation or sun effect on buildings has also been the subject of recent investigation. This research is of growing importance as the practice of cooling buildings in summer increases. A black roof surface normal to the sun's rays in summer will absorb heat, at noon with a clear sky, at the approximate rate of 300 Btu per square foot per hour. Due to the variation in the rate of solar radiation received, the heat capacity of the roof construction, the amount of heat radiated and dissipated to the air by convection, the amount of heat actually entering the building is naturally considerably less than the amount absorbed. An estimate of the amount entering the building has become an item to be allowed for in all calculations involving the amount of refrigeration required in dehumidification problems.

The amount of solar energy in the form of heat that a black body will absorb is the equivalent of approximately 5000 hp. per acre in the tropics. It is not to be wondered that physicists and engineers have sought for over a century to efficiently transform this vast source of energy into mechanical work.

The Claude-Boucherot experiments on the north coast of Cuba, which will attempt to utilize on a commercial scale the temperature difference of approximately 40 F between the surface water of the Gulf Stream and from a depth of approximately 2000 ft is being watched with great interest by engineers.

Research has provided friction coefficients for various heat conveying media in pipes and ducts so that it is possible accurately to predict the head lost and consequently the power consumed to convey steam, air and water under all operating conditions. Recent research of Prof. F. E. Giesecke has firmly established the friction coefficients for gravity flow hot water piping.

Effective Heat Output of Radiators

Engineers have recently been giving some thought to the so-called *effective warming* of direct radiation. Research has developed the fact that certain types of radiator covers or enclosures, in medium size rooms at least, will produce the same room air temperature at say 2 ft 6 in. above the floor with less steam consumption than an unenclosed radiator. This is due to the fact that the ceiling temperature is less with the cover or enclosure with a consequent smaller heat loss through the ceiling and upper sections of the side walls. Some engineers have suggested that direct radiators be rated on the basis of effective warming, rather than on the usual method of total heat emission. It is difficult to conceive how this could be satisfactorily accomplished.

Recent comparative tests on similar direct steam radiators of the same catalog rating furnished by several manufacturers have disclosed the fact that the heat emission is not the same and in some cases was found to be in excess of 15 per cent under the catalog rating, the catalog rating in all cases being based on a heat emission of 240 Btu per square foot per hour.

Panel Warming

The system of panel warming patented about 20 years ago by Professor Barker, an Englishman, and consisting of hot water pipe coils buried in the wall and floor construction is now apparently making some headway in England.

This system was recently installed in the British Embassy in Washington. With this method of heating comparatively large wall and floor areas are warmed to a maximum inside surface temperature of approximately 75 F by water circulated at a temperature of 120 F. The body heat loss, as previously mentioned, is regulated to a considerable extent by radiation to the cooler wall surfaces of the room. It is claimed with this system that a person normally clothed will feel comfortable in a room in which the air temperature is maintained at 60 F. This is, of course, impossible with the usual installation of direct radiation.

This method of heating apparently permits carrying

a considerably higher relative humidity in the air of the room, owing to the permissible lower dry-bulb temperature, for the same dew-point temperature as indicated by moisture precipitation on the glass of single windows, and is obviously a desirable feature not to be found in any other heating system. This system of heating combined with an air-conditioned ventilating system would appear to be an ideal arrangement, but a somewhat expensive combination to install and maintain.

Extended Surface Radiators

Non-ferrous extended-surface type concealed radiators are being used to some extent in homes of the better class and in some recent office buildings. Owing to the comparatively small mass of this type of radiation, the boiler is subjected to a smaller starting up load for the same length of time allowed for the warming up period, or conversely for the same starting up load on the boiler, the time for heating up the radiator is very materially reduced. For the same reason when automatic temperature control is employed this part of the equipment is evidently called upon to perform its function more frequently.

The manufacturers of cast-iron radiation reported sales of 139,000,000 ft for 1929, so that it cannot be said that this type of direct radiator has become entirely obsolete. Extended surface copper coil blast radiation has to a very large extent supplanted the cast-iron type, which for many years was considered standard in the blast field. Compactness, less labor required for installation and less weight to be carried by the supporting structure of the heater have been the contributing factors for its wide adoption in this field of heating.

Air Conditioning

There has recently been placed on the market, by several manufacturers, a combination of floor type unit heater and humidifier, complete with temperature and humidity control, which is finding a ready market in many industries.

The past twenty years has witnessed the birth and progress of the air conditioning industry from an experiment to its present highly developed state. The development of this branch of the heating and ventilating industry owes its advancement principally to research covering the control of psychrometric changes in the atmosphere and the development of the Carrier rational psychrometric formula resulting from this research. Perhaps the greatest contribution made to the art during the past 50 years, may be said to be air conditioning. It has been responsible, in industry, for the improvement made in the quality of many manufactured products and in some industries it is essential to the process.

It is, in many cases, responsible for the acceleration of the process, the elimination of economic loss due to material waste and the reduced efficiency of the employes, as a result of uncontrolled temperature and relative humidity. In order to maintain a fairly constant relative humidity within a structure the year around, dehumidification is essential and requires refrigeration.

The successful cooling of theatres during the summer months and the control of temperature and relative humidity in assembly halls has required the correct solution of many new, varied and somewhat complicated problems. It is of interest to note that the recent large installations of air conditioning apparatus, serving the House and Senate in the National Capitol Building and many other installations, are regulated in conformity with the requirements of the A. S. H. V. E. comfort chart.

Apropos to the subject of theatre ventilation the following somewhat startling information is quoted from a recent issue of the Civic Players Magazine and was extracted from A Young Man's Guide, by Wm. A. Alcott, published in Boston in 1851:

"A German medical writer calculates," he writes, "that one in 150 of those who frequently attend theatres becomes diseased and dies from the impurity of the theatres." And especially those accustomed to sitting in the orchestra "because the carbonic acid which is formed by respiration is heavier than atmospheric air and accumulates near the floor."

Constant theatre goers, he says, are stricken with colds, rheumatism, pleurisy, fever and consumption. Lacking malice, Mr. Alcott is genuinely moved by the lamentable prospects; "It is painful to look around on a gay audience of 1500 persons," he writes wistfully, "and consider that ten of this number will die in consequence of breathing the bad air of the room so frequently."

It is growing more apparent that the comparatively simple process of warming the interior of a building does not fully conform to our present day standards of living and comfort. Measuring the great majority of systems now in use by these standards, how few installations could qualify as satisfactory.

Comfort and Health

There are some who apparently confuse comfort with health and have consequently criticized the engineers' standard for comfort. The comfort chart is not a complete measure of healthful atmospheric condition and the engineers have never made any such claim. It is obvious that one may feel quite comfortable and be asphyxiated. It is equally obvious, however, that a heating system should promote comfort with no detriment to our health. The air we breathe should naturally be comparatively free from dust, odors and bacteria.

The synthetic air chart devised by Dr. E. V. Hill, as a measure of what is termed ventilation, is the only yard stick in existence which takes into account many of the contributing factors now thought to constitute good, bad or indifferent ventilation. We delight in giving credit to our world renowned scientists in their apparent solution of the fundamental structure of the atom and consequently all matter. It appears strange, however, to us laymen, that scientists and physicians cannot tell us today just what the so-called vital characteristics are, of perhaps the most important and certainly most universally employed combination of elements which contribute to our existence.

A recent laboratory experiment made upon three groups of small animals disclosed the rather surprising fact that the group supplied with an accurately prepared mixture of pure oxygen and nitrogen in the proportion usually found in the outside air died within a few hours. Research work has been started at Yale University in co-operation with the A. S. H. V. E. Research Laboratory in an attempt to solve the *vital characteristic* mystery. Only future research can disclose the

facts upon which engineers may build into a ventilating system the required apparatus to produce an artificial atmosphere, which can be safely said to equal that supplied by nature in its so-called health giving qualities. It is not unreasonable to suppose that artificially created atmosphere will be employed in the future to prevent, treat and combat some of the ailments of mankind susceptible to treatment through the medium of the lungs.

The accumulated knowledge of this art cannot be said to be applied today in a complete form to the heating and ventilation of the home, apartment house, or office building. What today may seem a luxury, tomorrow frequently becomes a necessity. The American public have been educated through the medium of advertising to the employment of a great variety of so-called home comforts, consisting principally of electrical appliances of various kinds.

It would appear that the industry should be capable of developing for the small and medium sized home, something better than the same types of systems that have been installed for the past 50 years with comparatively little improvement and furthermore with little or no increased cost to the owner.

In the light of our present knowledge of this art, the central fan blast humidity controlled system, appears to offer an approach to a practical solution of the problem. This system with complete automatic control is now being installed in a large number of high grade homes with apparent satisfactory results. Public demand in the near future will undoubtedly bring about a very decided change in heating methods and along with these changes may be expected better business for this industry primarily due to the replacement of existing installations that have become obsolete.

A Factor in the Temperature of the Stratosphere[1]

By W. J. Humphreys[2], C.E., Ph.D.

WHEN we first heard, some thirty years ago, that the temperature of the air rather rapidly decreases with increase of height up to the level of the highest cirrus, or wispy, clouds, and from there on, as far as a balloon could carry a thermometer, remains practically constant, we just didn't believe it—not all of it. We accepted, of course, the first part of the statement to the effect that the greater the height the colder the air. We had gotten used to that from mountain climbing, and from the records kept by balloonists. Then, too, we had found a good physical reason why it should be so, or at any rate an essential part of that reason. It is this: Ascending air expands because the pressure on it grows less and less by the weight of the air left below, but it expands against the weight of the air that still is above it, and therefore does work. Now, to do work it must expend energy, and its only available energy for this purpose is its heat. Evidently, then, ascending air, expanding as it goes, and doing work at the expense of its own heat, must get colder and colder with increase of height. All this is in perfect accord with our laboratory experiments, and so we accepted the fact of the decrease of temperature with increase of height as a phenomenon which, if not entirely self-evident, at least is so easy to explain as scarcely to merit a passing thought.

That is where we made at least two mistakes. In the first place, even when it does occur it isn't half so easy to explain as we thought it was, and, in the second place, it doesn't occur at all in the high atmosphere. Of course the pressure continuously decreases with gain of level beyond the highest clouds, just as it does below them, and so asking us to believe that the temperature does not also decrease up there with increase of level just as it does in the cloud region was asking too much; it was contrary to our laboratory experience. However, after hundreds of records obtained by sounding balloons (small balloons carrying only registering instruments) had shown that immediately the uppermost cloud level is passed the temperature really is practically constant, why of course we had to accept it as a fact, and revise our explanations accordingly.

In the end it all came out simply enough. Our previous reasoning had been perfectly correct, but the premises were sadly deficient. We had left out of account the effects of radiation, and had set no limit to convection. Throughout all that portion of the atmosphere in which clouds of any kind occur, that is, from the surface up to the height of 6 to 7 miles, in middle latitudes, and 8 to 10 miles in tropical regions, there is decided convection—change in level of individual masses of air. The temperature, therefore, of each such mass does vary with height, and as the whole of this portion of the atmosphere is involved in this continuous vertical turn-over, so also does this temperature relation extend to its every portion. But beyond the clouds vertical convection, if it exists at all, is so slow as to be practically absent so far as temperature effects are concerned. Here no one portion of the air changes temperature with altitude for the good and sufficient reason that it neither rises nor falls. Heat is not added to it by compression, nor taken from it by expansion. It therefore comes to that particular temperature at which it can lose heat by radiation at exactly the same rate that it gains heat by absorption. From this level up the intensity of the radiation from the earth and atmosphere below is practically independent of height. That is why temperature up there also is independent of height. It doesn't change appreciably even from day to night, and so we infer that it is not much affected by sunshine directly.

Thus again we had come to that state of mental ease that goes with understanding. But the ease was of short duration. It soon was found that the upper air, the stratosphere, as scientists now call it, is coldest over equatorial regions and becomes gradually warmer with increase of latitude, the extreme difference being around 35 F—coldest over the warmest earth and warmest over the coldest earth. Here was a poser, and we are not through trying to explain it yet.

[1] From the *Journal of the Maryland Academy of Science*, January, 1930.
[2] Professor of Meteorological Physics, United States Weather Bureau.

How Comfort is Affected by Surface Temperatures and Insulation

By Paul D. Close[1], New York
MEMBER

The importance of a hitherto neglected factor in the design of a heating system and the maintenance of the proper atmospheric conditions in a building, has recently been emphasized by Professors Willard and Kratz of the University of Illinois.[2] This factor relates to the radiation of heat from the body to cold surfaces, producing a feeling of chilliness, even under conditions which ordinarily would be considered satisfactory from the comfort standpoint. This paper contains a discussion of bodily comfort, including the effect of the heat emitted by radiation, as well as a mathematical analysis of surface temperatures, the direct relation of insulation thereto, and the consequent indirect relation of insulation to comfort.

UNTIL a few years ago it was thought that the comfort of our atmospheric environment depended solely on the temperature registered by the dry-bulb thermometer. However, the investigations at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING EGINEERS,[3] and at the *Harvard School of Public Health,* have disclosed the fact that the sensation of comfort depends not only on the dry-bulb temperature of the air, but also on the amount of moisture it contains and the rate of air motion. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content, and air motion makes any moderate condition feel cooler. These tests were used as the basis of an experimentally determined scale, known as *effective temperature,* which is a true measure or index of a person's feeling of warmth in all combinations of temperature, humidity and air motion.

Relation of Wall and Ceiling Temperatures to Comfort

As pointed out by Professors Willard and Kratz, there is also another factor to be considered in many cases, namely, the temperature of surrounding objects, particularly the wall, window and ceiling surfaces, if adjoining unheated spaces. In the process of metabolism, heat is continually being radiated from the body to the colder surrounding surfaces. Since, according to the Stefan-Boltzman law, the rate at which heat is emitted from the body by radiation is proportional to the difference of the fourth powers of the absolute temperatures, the effective temperature scale applies only when the temperature of the surrounding objects is at or near the surface temperature of the skin, and does not take into consideration any appreciable difference which may exist between the wall and skin temperatures.

Valuable information on the subject of body radiation is contained in *Smithsonian Institution* Publication, No. 2980, by L. B. Aldrich, entitled, A Study of Body Radiation. Reference is made to experiments conducted in the Nutrition Laboratory of *Carnegie Institution* in Boston, and in the *Smithsonian Institution* in Washington. The Smithsonian experiments were made at the request of the *New York Commission on Ventilation,* and the results of these experiments, as well as those conducted at the *Carnegie Institution,* are summarized in the aforementioned publication as follows:

1. The radiation from the skin and clothing is approximately that of a *black body* or perfect radiator.

2. A cloth-covered, vertical, cylindrical calorimeter at body temperature loses in still air 60 per cent by radiation and 40 per cent by convection. A similar horizontal calorimeter loses 54 per cent by radiation and 46 per cent by convection. The human body convection loss is probably similar to this, that is, the convection loss is roughly one third less than the radiation loss, in still air and normal room temperatures.

3. Increasing air motion rapidly decreases the percentage radiation loss and increases the convectional.

4. Total body radiation similarly decreases with air motion.

5. Increase in room temperature (which also means increase in wall temperature) produces a progressive lowering of radiation loss.

6. Keeping room and wall temperatures unchanged, the temperature of skin and clothing decreases with increasing air motion, the decrease being greatest on the side facing the wind and about one half as great on the side away from the wind. The clothing temperature drop on the side towards the wind is about one third greater than the corresponding skin temperature drop.

7. At normal indoor temperature, in still air and with the subject normally clothed and at rest, body heat losses are distributed as follows:

Evaporation of water	24 per cent
Radiation	46 per cent
Convection	30 per cent

8. The air temperature falls to room temperature very rapidly as the distance from the body increases. That is, there is a steep temperature gradient in the first ½ in. or so from the body surface. With the thermoelement about 12 in. away, no effect of the presence of the body could be detected.

9. The radiation loss from a nude subject is about twice as great for a room temperature of 59 F as it is for a room temperature of 79 F.

[1] Technical Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
[2] See *Wall Surface Temperatures* by Willard and Kratz, April 1930, A.S.H.V.E. JOURNAL.
[3] See A.S.H.V.E. TRANSACTIONS, Vols. 28 to 34 inclusive.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILAING ENGINEERS, Minneapolis, Minn., June, 1930.

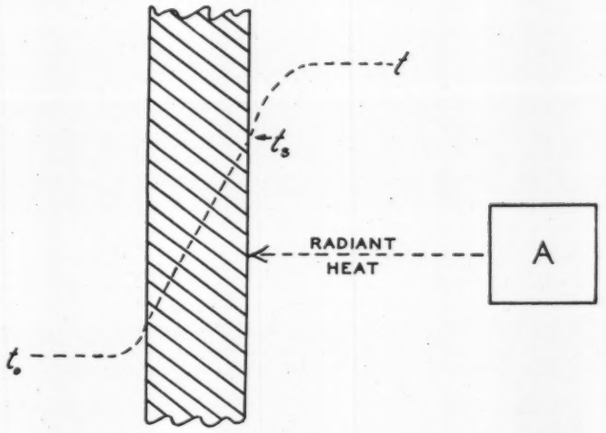


FIG. 1—SECTION THROUGH WALL SHOWING TEMPERATURE GRADIENT

10. Normal fluctuations in humidity indoors produce negligible effect upon the radiation loss. Were the air of the room exceedingly dry, changes might be noticeable.

Particular attention is called to Item 5 to the effect that an increase in wall temperature produces a decrease in the radiation loss. Consequently, if the wall and ceiling temperatures are increased sufficiently, a lower room temperature is permissible, provided the relative amount of window surface is not excessive. This is one of the principal features of the panel system of heating extensively in use in England,[4] in which the heating pipes are embedded in the walls or ceiling, and hot water forced through them to maintain the surfaces at a temperature of about 75 F. The increased surface temperature decreases the heat loss from the body by radiation, and thereby permits the maintenance of a lower room temperature, which results in fuel economy.

Attention is also called to Item 9 to the effect that the radiation loss was about twice as great for a room temperature of 59 F as it was for a room temperature of 79 F. This result was obtained from experiments conducted by Dr. C. G. Abbott and Dr. F. G. Benedict, in the *Carnegie Institution* in Boston. That the radiation loss increased with the decrease in the room temperature was no doubt due primarily to the fact that the lower room temperature resulted in a lower wall temperature, with a consequent increase in the difference between the wall, window and skin temperatures. Of course, lowering the room temperature would also tend to lower the skin surface temperature, but on a cold day the wall surface temperature apparently decreases to a greater extent than the skin surface temperature for a given reduction in the inside air temperature. According to Du Bois, when the room temperature is lowered, one changes one's integument into a *suit of clothes* and restores the zone where the blood is cooled from the skin to a level some difference below the surface.

If the temperature of the interior surfaces is sufficiently lower than the body surface temperature, the feeling of chilliness will result even though the normal effective temperature is maintained. This, as previously stated, is due to the increase in the loss of heat from the body by radiation. Since the rate at which heat is given off from the body by radiation increases as the temperature difference between the body surface and the wall and ceiling surfaces increases, the sensation of chilliness will increase to some extent as the wall surface temperature decreases, unless the increased loss of bodily heat by radiation is counterbalanced by an increase in the room temperature. Likewise, the less the heat resistance of the wall, the lower the interior wall surface temperature for the same outside temperature, and the greater will be the sensation of chilliness for the same effective temperature. Consequently, if the walls of a building are poorly insulated, or if they contain a large percentage of window surface, the effective temperature must be increased in cold weather to offset the increased loss of heat from the body.

Effect of Insulation on Surface Temperature

Inasmuch as the wall surface temperature is a function of the overall heat resistance of the wall, the addition of a given thickness of insulation will materially increase the surface temperature, and permit the maintenance of the normal effective temperature, provided the relative amount of window surface is small. Under the circumstances, the insulation will result in a double heat saving, *first*, that due to the lower temperature difference, and *second*, that due to the retarding of the rate of heat transfer through the wall.

A study of the variation in the wall surface temperature with the various factors affecting this temperature may be of interest. The temperature of the interior surface of a wall (see Fig. 1) may be determined from the following formula (see next page):

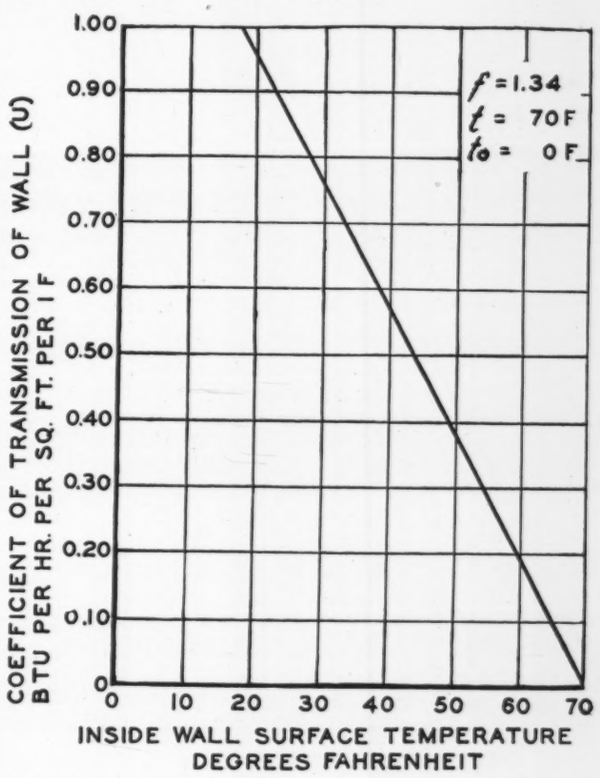


FIG. 2—CURVE SHOWING RELATION BETWEEN INTERIOR SURFACE TEMPERATURE OF WALL AND COEFFICIENT OF TRANSMISSION OF WALL WHEN f, t AND t_o ARE CONSTANT AND HAVE THE VALUES INDICATED

[4] See Panel Warming, by L. J. Fowler, A.S.H.V.E. JOURNAL (*Heating, Piping and Air Conditioning*), January, 1930.

$$t_s = t - \frac{U}{f}(t - t_o) \qquad (1)$$

where—

t_s = temperature in degrees Fahrenheit of the interior wall surface

t = inside air temperature

t_o = outside air temperature

U = coefficient of transmission of wall in Btu per hour per square foot per degree Fahrenheit difference in temperature

f = film conductance of interior wall surface expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air in the room and the wall surface.

From formula (1) it is apparent that the wall surface temperature is a function of the inside and outside air temperatures, the coefficient of heat transmission of the wall and the interior surface or film conductance of the wall. Sun effect also has a bearing on the inside surface temperature, but is neglected in this analysis. Consider an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, the coefficient of transmission of which (according to the A.S.H.V.E. Guide 1930) is 0.356 Btu per hour per square foot per degree Fahrenheit difference in temperature. If the inside and outside temperatures are 70 F and 0 F, respectively, and the surface conductance is 1.34, the temperature of the interior wall surface will be

$$t_s = 70 - \frac{0.356}{1.34}(70 - 0) = 51.4 \text{ F}$$

Variation in Surface Temperature with Various Factors

The curve (Fig. 2) illustrates the variation in the wall surface temperature with the coefficient of transmission (U) of a wall, assuming the inside and outside air temperatures to be constant at 70 and 0, respectively, and the surface conductance to be 1.34 (average value used in The A.S.H.V.E. Guide 1930) It will be noted that if U has a value of 1.00—which is the approximate coefficient of a sheet metal wall (flat surface) when exposed to a wind velocity of 15 miles per hour—that the wall surface temperature is only 17.8 F. If the curve were extended,

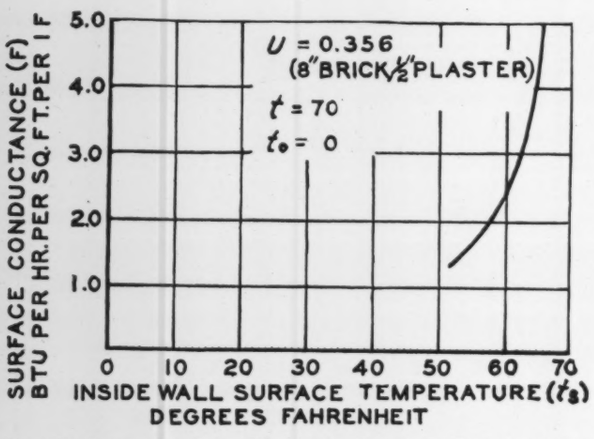


Fig. 3—Curve Showing Relation Between Interior Surface Temperature of Wall and Interior Surface Conductance of Wall When U, t and t_o Are Constant

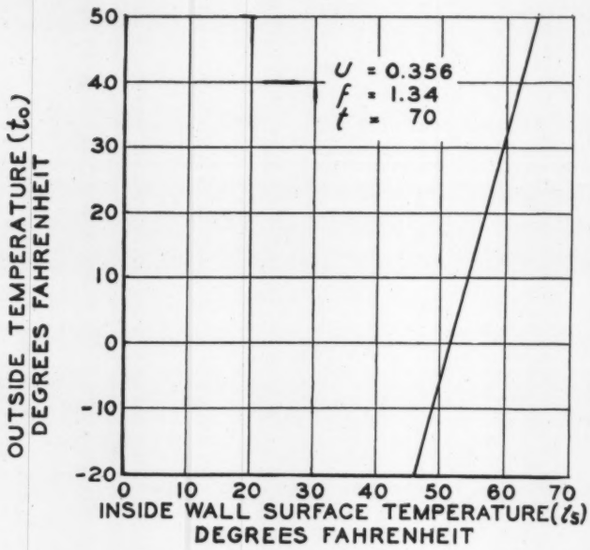


Fig. 4—Curve Showing Relation Between Interior Surface Temperature of Wall and Outside Temperature When U, f and t are Constant and Have the Values Indicated

t_s would be equal to t_o when $U = f$ or 1.34. As the coefficient of the wall (U) approaches 0, the inside surface temperature approaches the inside air temperature (t) which in this case is 70 F. If the 8-in. brick wall under consideration is insulated with 1-in. of rigid insulation having a conductivity of 0.33, so that the coefficient is reduced to 0.148, the inside surface temperature would be 62.3 F for the conditions involved. Hence, the insulation in this case, together with the air space resulting from the installation thereof, would increase the surface temperature 10.9 F on a zero day, and would therefore have a marked effect upon the degree of comfort of the room, because of the resulting reduction in the loss of heat by radiation. An increase in the surface conductance increases the surface temperature when the other factors remain constant, as will be seen from Fig. 3. For a still air conductance of 1.34, an 8-in. brick wall with ½-in. of plaster, will have a surface temperature of 51.4 F as previously stated. If the velocity of air passing over the surface is increased, the surface conductance will increase. At a conductance of 2.00, which would result from a slight wind movement, the surface tem-

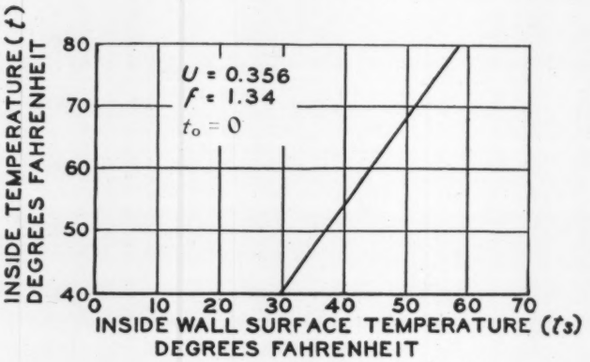


Fig. 5—Curve Showing Relation Between Interior Surface Temperature of Wall and Inside Temperature When U, f and t_o Are Constant and Have the Values Indicated

perature will be increased 6.1 degrees to 57.5 F, assuming the inside and outside temperatures to be 70 F and 0 F, respectively. Although the value of U will increase slightly with an increase in the surface conductance, it is assumed constant in this case, as this assumption will not appreciably alter the results. If the velocity of the air passing over the interior surface is increased to about 15 miles per hour, the surface conductance will be 4.02, assuming the 3 to 1 ratio used in THE GUIDE 1930, and

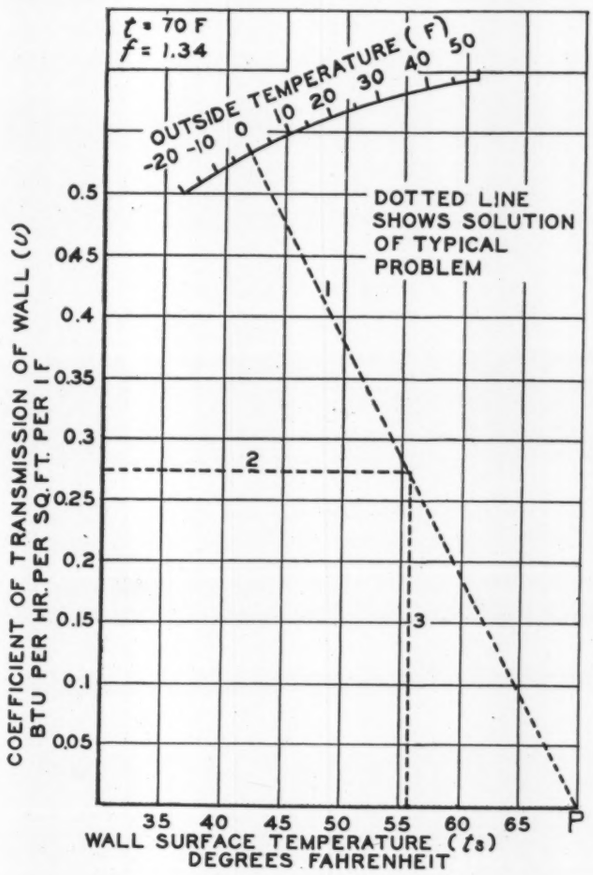


FIG. 6—CHART FOR DETERMINING WALL SURFACE TEMPERATURE FOR VARIOUS OUTSIDE TEMPERATURES AND WALL COEFFICIENTS

the surface temperature will be about 63.8 F, an increase of 12.4 F over the still air condition.

It is apparent that an increase in the outside temperature will increase the interior surface temperature, whereas a decrease in the inside temperature will decrease the interior surface temperature. These relationships are shown graphically by Figs. 4 and 5. For the type of wall under consideration (8-in. brick, ½-in. plaster), the inside surface temperature increases from 46 F to 64.7 F when the outside temperature is 50 F, assuming that the inside temperature remains constant at 70 F. If the inside air temperature varies, the wall surface temperature will increase from 29.4 F when the inside temperature is 40 F, to 58.8 F when the inside temperature is 80 F, assuming that the outside temperature remains constant at 0 F.

The inside temperature is usually about 70 F and the surface conductance may be considered constant for walls and ceilings, and having a value of 1.34. Hence, in most cases the only variables to be taken into consideration are the outside temperature and the coefficient of transmission of the wall. The chart (Fig. 6) may therefore be used for estimating the wall surface temperature for any combination of these two variables.

Windows

Window surface temperatures for single glass are considerably lower than wall surface temperatures on a cold day. Hence, if the exterior walls of a room contain an excessive amount of glass, double or triple windows will materially reduce the loss of heat from the body by radiation. The coefficient of transmission (U) of single glass, based on a wind exposure of 15 miles per hour, is 1.13 (A.S.H.V.E. GUIDE 1930) and the still air surface conductance (f) is 1.50. Hence, on a zero day, with an inside temperature of 70 F, the glass surface temperature will be—

$$t_s = 70 - \frac{1.13}{1.50}(70 - 0) = 17.3 \text{ F}$$

With double glass, the surface temperature will be 49 F, based on a coefficient of transmission of 0.45, and with triple glass the surface temperature will be 56.9 F, based on a coefficient of 0.281. Therefore, double glass used in place of single glass will increase the surface temperature on a zero day to the extent of 31.7, whereas triple glass will increase the surface temperature 39.6 F. Consequently, the effect of double or triple glass in the case of windows is even more pronounced than the effect of insulation in the case of walls; hence the importance of using double or triple glass in a room having a large amount of window surface.

Insulation and Heat Loss by Radiation

The Stephan-Boltzman radiation law may readily be applied to a typical problem involving the change in wall surface temperature resulting from the installation of a given thickness of insulation. The metabolism of the human body is somewhat complicated. Heat is given off not only by radiation but also by convection and evaporation of water. The physiological processes are continually striving to adjust themselves to the surrounding atmospheric environment in order to equalize the heat produced and the heat given off from the body, and thereby to maintain bodily comfort. Hence, any change in the loss of heat by radiation may also involve a change in the loss by convection and evaporation as well as a change in the temperature of the exposed skin. These changes do not readily lend themselves to mathematical analysis, and for the sake of simplifying the calculations, an inanimate object instead of the human body will be considered in the following problem.

Assume the surface of an object (A, Fig. 1) to be maintained by electrical means or otherwise at a constant temperature of 90 F in a room temperature of 70 F. Except for the changes in physiological reactions, this object may be considered analogous to a human body in this analysis. The problem is to determine the decrease in the rate of heat emission by radiation from this object to the exterior walls on a zero day resulting from the installation of a 1-in. thickness of a flexible insulation to the brick veneer frame walls of the room.

The coefficient of transmission of the uninsulated wall is 0.247 (A.S.H.V.E. GUIDE 1930), and the surface tem-

perature, according to formula (1) is approximately 57 F. The absolute temperature, therefore, is 57 + 460 or 517 degrees. The coefficient of transmission of the insulated wall is 0.116, the surface temperature, 64 F, and the absolute temperature, 524 degrees. The decrease in the loss of heat by radiation, according to the Stephan-Boltzman law:

$$= \frac{[(5.50)^4 - (5.17)^4] - [(5.50)^4 - (5.24)^4]}{(5.50)^4 - (5.17)^4}$$

$$= \frac{200 - 160}{200} = 0.20 \text{ or } 20 \text{ per cent}$$

Therefore, under these conditions, the insulation would reduce the rate at which radiant heat is absorbed by the wall to the extent of 20 per cent. While this analysis would not apply exactly to a human being, the result is indicative in a measure of the effect of the insulation on comfort. An increase in the loss of heat from the body by radiation would tend to lower the skin temperature. If the temperature of the object were only 75 F, the insulation would reduce the rate of heat loss to the wall by radiation, 39.2 per cent.

Consider the problem from another angle. To what degree would it be necessary to raise the inside air temperature in the case of the uninsulated wall (disregarding windows) in order to obtain the same rate of heat emission by radiation as in the case of the insulated wall?

In solving this problem, it is merely necessary to equate the radiation emission (Q) in the two cases, and solve for t as follows:

$$Q = D\left[\left(\frac{460 + 90}{100}\right)^4 - \left(\frac{460 + t_s}{100}\right)^4\right]$$

$$= D\left[\left(\frac{460 + 90}{100}\right)^4 - \left(\frac{460 + 64}{100}\right)^4\right]$$

where—

D = radiation constant
t_s = wall surface temperature
= 64 F

From formula (1)—

$$t_s = t - \frac{0.247}{1.34}(t - 0)$$

$$= 0.816\ t \text{ and}$$

$$t = 78.5 \text{ F}$$

It would therefore be necessary to increase the room temperature (t) from 70 F, in the case of the insulated wall, to 78.5 F, in the case of the uninsulated wall, to obtain the same heat loss from A (Fig. 1) by radiation. Of course, it probably would not be necessary to increase the air temperature to this extent to produce the same bodily comfort—in fact, this dry bulb temperature without a correspondingly low wet-bulb temperature would ordinarily produce a feeling of lassitude or discomfort—but the result gives a hint of the importance of insulation from the standpoint of winter comfort, and the necessity of increasing the temperature in the zone of occupancy of an uninsulated room to offset the increase in the loss of heat from the body by radiation.

Sun Effect on Buildings

Sun effect, or the absorption of solar radiation in the summer, by the surfaces of a building exposed to the sun, has an important bearing on the capacity of the refrigerating equipment required for air conditioning. These surfaces may be the exterior walls or the roof of the building, or interior surfaces which receive the sun's rays through window glass.

Heat Transfer Through Exterior Walls and Roofs

When equilibrium has been established, and the rate of heat flow is therefore constant, the heat transmitted per hour through a wall or roof is equal to the product of the conductance of the structure from the exterior surface to the interior air and the difference between the exterior surface temperature and the interior air temperature. This relation is expressed as follows:

$$H = C_1 (t_r - t) \qquad (1)$$

where

H = Heat transmitted through wall or roof, Btu per square foot, per hour.
C_1 = Conductance of wall or roof between exterior surface and interior air, Btu per square foot, per hour, per degree Fahrenheit difference in temperature.
t_r = Exterior surface temperature, degrees Fahrenheit.
t = Interior air temperature, degrees Fahrenheit.

Under actual conditions, the heat flow due to sun effect is almost never in equilibrium because of the heat capacity of the wall or roof structure. The exterior surface temperature of a wall or roof will usually approach the maximum for the day shortly after midday. On the other hand, the temperature of the interior surfaces of the building will lag, due to the heat capacity of the structure, and consequently the maximum interior surface temperature that would exist, if the heat flow were in equilibrium, may not be reached until some time after the maximum exterior surface tmperature has been attained. Since at that time the exterior surface temperature has diminished, the maximum interior surface temperature (based on constant flow, and the maximum exterior surface temperature) will not actually be reached, and consequently the heat to be absorbed will be somewhat less than would be required according to Equation (1).

For materials of low heat capacity, the lag will be small, and the error resulting from the use of Equation (1) will be small. In any event, the result will be on the safe side. For materials of high heat capacity, a correction for the temperature lag of the interior surfaces should be made, although at present data are lacking for accurately evaluating the effect of heat capacity on the refrigerating requirements of air-conditioned buildings. If the maximum interior surface temperature is known, the heat transfer may be estimated by means of the following formula:

$$H = (t_s - t) \times f_i \qquad (2)$$

where

t_s = Maximum interior surface temperature.
f_i = Interior surface conductance.—*P. D. C.*

Centrifugal Fans

By G. S. Wilson[1], Seattle, Wash.
NON-MEMBER

FOR HEATING and ventilating service, large quantities of air are required to be moved against small pressures. The centrifugal fan was designed for this purpose. A brief historical sketch of its experimental development is of interest.

Among the earliest experiments on centrifugal fans were the investigations of Hammersley Herman and William Gilbert. Their object was: (*a*) "to determine the best shape of fan blade and fan case, in order to secure a minimum expenditure of power when producing any given output of air, *i.e.*, to find the best type of fan; (*b*) the standard type being selected to obtain data whereby the proper diameter of the standard fan and its most economical speed could be determined for any required output of air at a given pressure."

The best type of fan is still being sought, but the experimental data obtained by Herman and Gilbert, furnished desirable information on fan operating characteristics and velocity distribution across the fan outlet. The authors verified the following laws for constant resistance:

1. Volume of air discharge varies as the speed.
2. Pressure varies as the speed squared.
3. Horsepower varies as the speed cubed.

The next experimenter of note was Bryan Donkin who made a large number of tests in which the number and shape of the vanes were changed. Great care was taken in measuring the air volume and although the tests were made on small fans he is given considerable credit for his results.

The Prussian Mining Commission tests of 1884 were of interest in that they devoted considerable time in determining the possibilities of anemometers, Pitot tubes and thin plate orifices for the measurement of air velocities and volumes. They also studied friction losses as affected by length and diameter of discharge pipe.

In 1892 the Belgian Commission conducted tests on a number of mine fans to determine (1) volume of air discharged, (2) efficiency, (3) safety and (4) cost.

Since this date various experimenters have added data on blade shapes and passages, discharge and intake angles, shape of fan housings, etc.

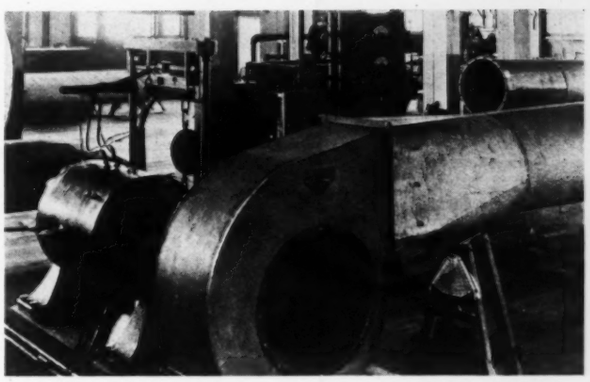


FIG. 1—ARRANGEMENT OF THE EQUIPMENT IN THE MECHANICAL ENGINEERING LABORATORY OF THE UNIVERSITY OF WASHINGTON

In the mechanical engineering laboratory at the *University of Washington,* complete equipment for fan testing is available. This includes one small and one large electric dynamometer for measuring power input, Pitot tubes and multiplying gages for measuring static and velocity pressures, wet- and dry-bulb thermometers, continuous counters and all accessories for conducting tests in accordance with the standard test code as provided by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Fig. 1 shows the arrangement of equipment in the *University of Washington* Mechanical Engineering Laboratory.

Fig. 2 shows fan operating characteristics. The fan is operated at constant speed and the discharge is varied from 0 to maximum, in 8 or 10 steps, by changing the

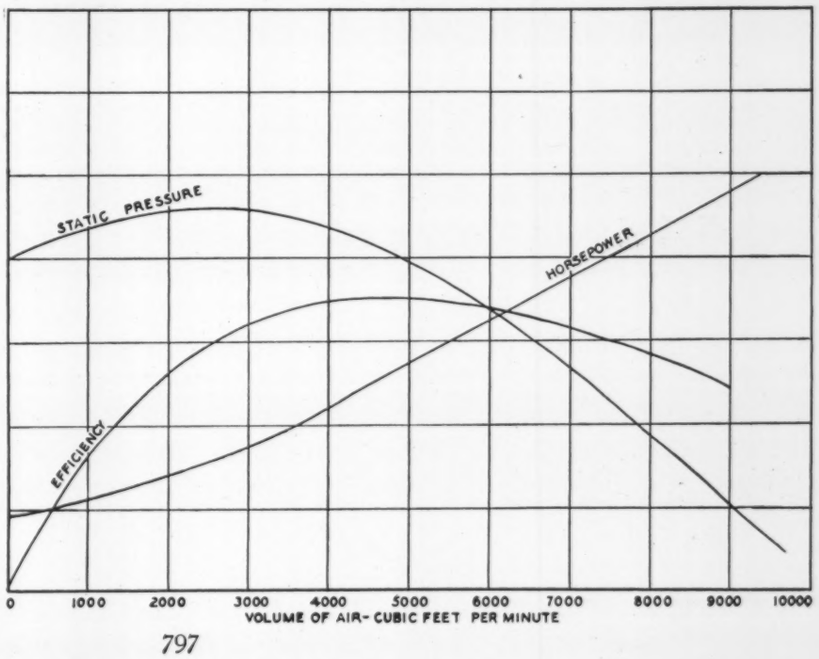


FIG. 2—CURVES SHOWING FAN OPERATING CHARACTERISTICS

[1] Professor of Mechanical Engineering, University of Washington.

Paper presented at the April 1930 Meeting of Pacific Northwest Chapter of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

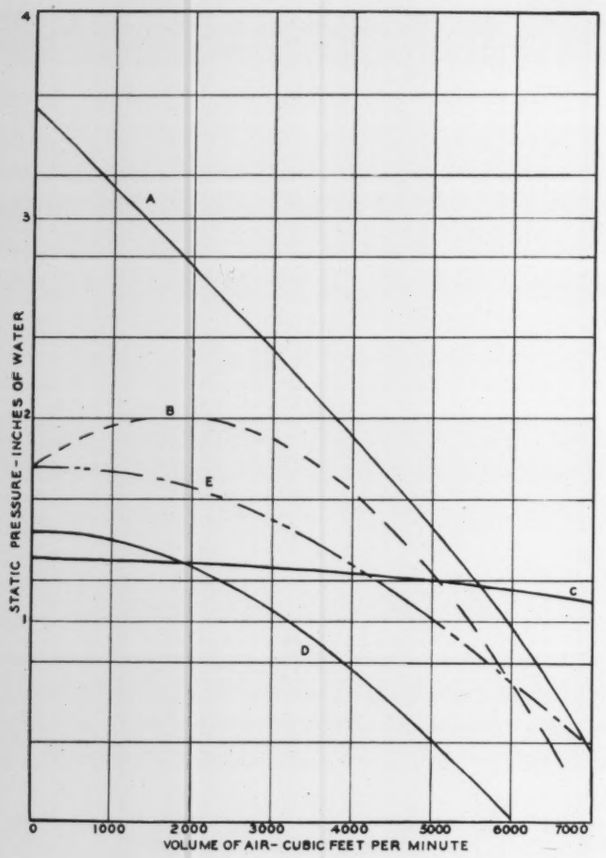


FIG. 3—STATIC PRESSURE CHARACTERISTICS OF A FAN

shutter on the outlet of the discharge pipe. Observations are made at each step and the results are shown on the graph.

W. L. Dudley and the writer have made several tests on centrifugal fans. The studies included the effect of blade passages, blade shapes, depth of blade, number of blades, inlet angles, discharge angles, size and shape of housing, variation of velocity across the fan outlet and volumetric efficiency of fan. A great amount of useful information was obtained. These data, together with the great store of similar experimental data that are available, make it possible either to select or design a fan suitable for a given installation.

The heating and ventilating engineer designs the ducts, finds out the demands to be made upon the fan and with the assistance of this experimental information, selects the unit that will serve to the best advantage.

Members of the Pacific Northwest Chapter of the A. S. H. V. E. are largely concerned with the selection of fans for definite purposes and the experimental data assist in the solution of the problem. No doubt many feel the need of a theoretical analysis of a given type of fan that will make it possible to predict, within reasonable limits, its operating characteristics over a considerable range.

The static pressure characteristics as shown in Fig. 3 will illustrate this point. The probable static pressure curve is calculated on the basis of the fan being operated at constant speed. The abscissa is air volume.

Curve *A* shows the possible static pressure when using one authority's method. The curve is correct but it shows the theoretical possible pressure and allows for no losses.

Curve *B* shows another method in which allowance is made for certain losses.

Curve *C* is still another method in use for calculating pressures.

Curve *D* is the static pressure obtained by actual test. It will be noted that curve *D* varies considerably from the prediction made by use of methods *A*, *B* and *C*.

Curve *E* is plotted by using a tentative expression worked out by Mr. Dudley and the writer. There is still a variation from the actual static pressure but the curve has the merit of being at least of the same general shape. The problem is not solved but progress is being made.

By-Product Coking

In a recent radio address, Charles J. Ramsburg, vice-president of the Koppers Company, made the following statement:

"Not a great many years ago we considered coal as a fuel only—a fuel which produced soot and smoke, damaging our homes and stealing our life-giving sunshine. Today the by-product coke oven makes coal an invaluable source of raw materials and at the same time gives us coke, that hard gray fuel without which iron and steel production by present methods and quantities would be impossible.

"For househeating, coke is an ideal fuel, clean to handle, clean to burn. Each year 100,000 homes are added to the list of those who realize the obvious benefits which accrue both to the individual and to his community when smokeless coke is used.

"Coal gas, the fuel de luxe for automatically heating homes without labor and without worry, for baking the lightest breads and the finest pastries, and for producing in industry the highest grade metal products, is simultaneously produced with its twin fuel coke. The uses of gas are myriad. Massive gas holders, familiar in every city, are the bins in which is stored this peerless fuel ready to serve at your command.

"Ammonia, a product of the coke oven is, in time of war, the base of the most powerful munitions. When war is over it turns its energies back to agricultural pursuits, entering into the composition of practically all fertilizers. Ammonia is the base of all large-scale refrigeration and familiar in the home as a cleanser. It is also used in multitudinous chemical industries.

"Benzol, the mother of a rapidly growing chemical family of over 100,000 members, is recovered in by-product coking. This family includes toluol, the base of the much-discussed T.N.T. of war-time, but now an important ingredient of paint for your motor car and of numerous other lacquers and varnishes. The same family includes a long list of dyes and drugs which have made the world brighter and safer in our generation.

"Tar, that black, viscous liquid obtained from by-product coking, is a willing slave of war and commerce. From it are produced explosives to wreck mighty fortresses, to sink majestic ships, and to mine the coal from which it is produced. From it are obtained the road materials with which thousands of miles of highways are built and resurfaced with non-skidding qualities, making it possible to motor in safety and dustless comfort."

Of Interest to Members

President Harding to Visit Western Chapter in October

During the month of October, Pres. L. A. Harding is scheduled to speak at meetings of the Society's Chapters in Chicago, St. Louis, Kansas City and Milwaukee. The subject of his talk will be Utilization of the Sun's Energy.

Following this in November and December he will visit Michigan Chapter, Massachusetts Chapter, New York and Philadelphia Chapters, on dates to be announced later.

The Chapters are making special arrangements to entertain President Harding and the Officers and Council Members who will be present at these meetings.

August Council Meeting in Buffalo

Pres. L. A. Harding called a meeting of the Council at the Hotel Statler, Buffalo, N. Y., on August 11 and the following members were present: D. S. Boyden, Boston, W. H. Carrier, Newark, J. D. Cassell, Philadelphia, Roswell Farnham, and C. W. Farrar, Buffalo, F. C. McIntosh, Pittsburgh, F. B. Rowley, Minneapolis and A. V. Hutchinson, Secretary, New York.

An important matter of business coming before this meeting was the nomination of members to serve on the Committee on Research and the five selected for a three-year term were C. A. Booth, Buffalo, D. S. Boyden, Boston, F. B. Howell, New York, Walter Klie, Cleveland and F. B. Rowley, Minneapolis.

Upon the suggestion of the Committee on Research a set of A. S. H. V. E. Transactions was presented to the Library of the U. S. Bureau of Mines, Pittsburgh Experiment Station.

It was decided to print the A. S. H. V. E. Transactions Index and make it available to the public at a nominal price.

Acceptance of the resignation of T. J. Duffield, Executive Secretary for the Committee on Research, by the Committee was reported to the Council and for the remainder of the year it was decided that the promotional work already started would be carried on by the Chairman and the Director of Research.

A considerable portion of the Council Meeting was devoted to a discussion of methods of financing and operating the Society's Research activities.

To Specialize in Consulting Engineering

C. P. Lichty, president of the C. P. Lichty Engineering Co., Inc., 1216 Martin Building, Birmingham, Ala., announced that the firm in the future will devote its efforts entirely to consulting engineering, specializing in heating and ventilating, electrical refrigeration, power and sanitary services.

J. S. Webb, New York Manager for Herman Nelson

Announcement has just been made of the appointment of J. S. Webb as manager of the New York branch of the Herman Nelson Corp. with offices at 25 West 43rd St. For a number of years, Mr. Webb has been with the Herman Nelson Corp. in the New England territory with headquarters in Boston.

His many friends and associates in the Massachusetts Chapter of the Society, which he has been serving as Secretary, regret that he is leaving Boston, but wish him great success in his new field of operations.

J. C. Fitts Elected Secretary H. P. C. N. A.

At the 41st Annual Meeting of the *Heating and Piping Contractors National Association*, which was held at the Hotel New Yorker, New York City, June 16 to 19, 1930, J. C. Fitts was elected Secretary to succeed H. B. Gombers, who was appointed Secretary-Emeritus.

Mr. Fitts, who has been with the Association since 1919, was graduated from Princeton University, Magna cum Laude, with the class of 1912, receiving his A. B. degree.

He then became principal of the Union Street School at Ridgewood, N. J., remaining there until 1916, when he entered the service of the U. S. A.

J. C. Fitts

During his service in the Army he held various Commissions, some of which are, First Lieutenant of Infantry, First Lieutenant of Engineers, and Captain of Engineers.

Since his affiliation with the *Heating and Piping Contractors National Association* in 1919 he has been secretary of the Committee on Standards, and until 1929, was secretary of the Committee on Certified Heating. He was also chairman of the Board of Directors of the Association for Correlating Thermal Research from 1928 to 1930.

Mr. Fitts is also the editor of the "Official Bulletin," which is published by the Association.

W. I. Lawson Joins New York Blower Co.

The New York Blower Co., Chicago, Ill., manufacturers of unit heaters, fans, blowers, etc., announces that William I. Lawson, who has been engaged in this industry for a number of years and who has a wide acquaintance in the heating and ventilating field, is now associated with this company as sales manager.

Mr. Lawson will be located at the general offices of the company, 32nd St. and Shields Ave., Chicago.

Southern California Members Meet

At a dinner meeting on July 29 in the University Club, Los Angeles, 53 men were present. The charter for the Southern California Chapter granted by the Council of the A. S. H. V. E. was read to the members present, as well as greetings from Pres. L. A. Harding, Past President Thornton Lewis and Secretary A. V. Hutchinson.

A proposed Constitution was presented for discussion and after hearing reports of committees on membership, programs and meeting arrangements, it was voted that the temporary officers, O. W. Ott and H. B. Keeling hold office until a Constitution and By-Laws could be adopted and permanent officers elected.

A talk on heating and ventilating practice in Australia was given by C. C. Sands of Sydney, Australia, and J. A. Armstrong of A. M. Byers Co. gave an interesting discussion of wrought iron pipe manufacture, illustrated by motion pictures.

The next meeting was scheduled for August 12 at the University Club for the purpose of adopting a Constitution and By-Laws.

1931 Annual Meeting Arrangements

The Pittsburgh Chapter is to be host to the Society for its 37th Annual Meeting at the William Penn Hotel, Pittsburgh, Pa., January 27 to 29, 1931, and the general arrangements for entertaining visitors will be directed by H. Lee Moore.

The General Committee is made up of the following men: J. E. McGinness, Publicity; F. C. McIntosh, Finance; W. W. Stevenson, Transportation; H. B. Orr, Reception; R. B. Stanger, Entertainment; F. A. Gunther, Banquet; and F. C. Houghten, Program.

It is planned to have an inspection of the Society's Laboratory and a technical session at the Bureau of Mines Experiment Station as well as several other novel features soon to be announced.

Special Society Committees Appointed

Several special committees have been appointed by Pres. L. A. Harding and are working industriously on the problems which have been presented to them. The Committee on Revision of Constitution and By-Laws consists of Thornton Lewis, Chairman, Philadelphia; W. T. Jones, Boston; and F. R. Still, New York.

The Committee on Chapter Relations appointed to secure a closer contact between the Society and Chapters, to provide for uniform operating regulations and to co-operate in the preparation of programs has R. H. Carpenter as Chairman, New York, assisted by H. H. Angus, Toronto, and F. C. McIntosh, Pittsburgh.

The Committee to revise the Code of Minimum Requirements for the Heating and Ventilation of Buildings is composed of Perry West, Newark, Chairman; C. A. Booth, Buffalo; W. H. Driscoll, Newark; H. P. Gant, Philadelphia; E. V. Hill, Chicago; and Prof. A. C. Willard, Urbana.

The Committee to Prepare the Rules of Award for the Anderson Medal are F. C. McIntosh, Pittsburgh, Chairman; J. D. Cassell, Philadelphia and Roswell Farnham, Buffalo.

Two Technical Committees to prepare Codes have been appointed as follows: Committee on Code for Testing and Rating Concealed Gravity Type Radiation, R. N. Trane, Chairman, E. H. Beling, W. F. Goodnow, Hugo Hutzel, A. P. Kratz, E. J. Vermere and O. G. Wendel; Committee on Code for Testing and Rating Unit Ventilators, John Howatt, Chairman, S. E. Dibble, G. E. Otis, L. D. Harnett, M. J. Callahan, A. J. Nesbitt and C. P. Bridges.

Engineers Available

The American Society of Heating and Ventilating Engineers, 51 Madison Avenue, New York, N. Y., has received the applications of several engineers who desire sales connections, or with engineering or contracting offices.

If further particulars about any of these men are desired they will be furnished promptly.

Sales

A 1 Experienced engineer with good acquaintance metropolitan district desires sales connection. Past experience has been in heating specialties, temperature control and unit ventilation work.

A 2 Young man, university graduate, desires selling connection in metropolitan area where contact is needed with engineers, architects and industrial plants; previous work specialized in ventilation.

A 3 Technically trained man of wide experience and good acquaintance in New York territory desires connection with manufacturer of heating specialties or unit heating and ventilating equipment where his previous experience in sales and engineering will be of value.

A 4 Technically trained man with acquaintance among engineers, architects and home owners in New York and Westchester desires a connection in sales work where special training in oil heating equipment might be of interest to a concern marketing such devices.

Engineering and Contracting

E-1 Man with good technical training and experience in estimating and sales work desires position with a large contracting concern.

E 2 Engineer with 10 years' experience with New York contracting and engineering firms desires position where his knowledge of design for heating and ventilating systems will be of value.

E 3 Experienced engineer desires connection in mechanical field where electrical and mechanical training can be used advantageously by a concern handling power, heating, ventilating, refrigeration and plumbing.

E 4 Man with both practical and selling experience in heating desires connection with contractor where his knowledge of estimating, designing, etc., can be applied.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 30 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 15, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ABBOUD, ALFRED, Alfred Abboud & Co., Inc., Boston, Mass. (*Advancement*).	J. F. Tuttle	
ANDERSON, EDWIN L., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	C. V. Fenn L. L. Lewis
ARMSTRONG, JOHN ALEXANDER, Mgr., A. M. Byers Co., Los Angeles, Calif.	O. W. Ott C. S. Anderson	H. B. Keeling H. A. Nelson
BENSON, JOHN C., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	F. P. Anderson L. S. O'Bannon
BYNUM, OTIS WILLIAM, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	L. L. Lewis J. B. Merrill
COBURN, LUTHER F., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	C. H. B. Hotchkiss L. L. Lewis
DARBY, MARION HENRY, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	H. W. Worsham L. L. Lewis
DOWLING, JOSEPH MICHAEL, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	Walter Grant L. L. Lewis
EDGAR, JAMES STRUTHERS, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	L. L. Lewis R. L. Jones
FITTS, JOSEPH C., Secy., Heating & Piping Contractors National Association, New York, N. Y.	H. M. Hart R. S. Franklin	F. C. Houghten H. G. Black
GILLHAM, JOHN N., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	F. P. Anderson L. S. O'Bannon
GRAY, RICHARD FULTON, Dist. Repres., Iron Fireman Manufacturing Co., Cleveland, Ohio.	W. C. Kammerer F. H. Morris	R. G. Davis H. M. Nobis
HASKETT, JOSEPH ELWOOD, Carrier Engineering Corp., Newwark, N. J.	W. G. Hillen W. H. Carrier	L. L. Lewis R. L. Jones
HIRST, JAMES NOBLE, York Heating & Ventilating Corp., Philadelphia, Pa.	J. V. Cavileer H. P. Gant	Clarence E. Scott C. D. Graham, Jr.
MANN, LEE BERNARD, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	L. L. Lewis W. H. Carrier
MEFFERT, GEORGE H., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	R. E. Cherne W. H. Carrier
PAGET, FRANCIS KING, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	L. L. Lewis H. W. Worsham
PERTAK, WILLIAM R., Allen & Billmyre Co., New York, N. Y.	L. F. Nordine W. I. Lawson	E. J. Ritchie M. W. Ehrlich
POWERS, LOWELL GABLE, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	H. W. Worsham L. L. Lewis
PULLIAM, WILLIAM GOLD, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	L. L. Lewis R. L. Jones
ROTH, CHARLES F., Vice-President, International Exposition Co., New York, N. Y.	Thornton Lewis H. P. Gant	J. D. Cassell W. H. Carrier
SCANLAN, CHESTER J., Designing Engr., The Trane Co., La-Crosse, Wis.	A. P. Kratz A. C. Willard	R. N. Trane R. H. Anderegg
SKIDMORE, JOHN GARDNER, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. I. Lyle	R. E. Cherne W. H. Carrier
SMITH, FRANK J., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	C. V. Fenn L. L. Lewis

SWENSON, JOHN EDWARD, Minneapolis Gas Light Co., Minneapolis, Minn.	M. H. Bjerken G. A. Dahlstrom	A. J. Huch C. E. Hill-Non member
TUPPER, GEORGE B., Airmaster Corp., Chicago, Ill.	W. Gardner, Jr. A. L. Weixel	G. C. Heck G. A. Muir
VAN METER, BENJAMIN F., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	F. P. Anderson L. S. O'Bannon
WALTON, LOUIS A., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	F. P. Anderson W. A. Grant
WARE, JAMES ELLIOT, JR., Carrier Engineering Corp., Newark, N. J.	W. G. Hillen J. B. Merrill	R. A. Regan H. W. Worsham
WITMER, CHARLES NEWCOMER, Carrier Engineering Corp., Newark, N. J.	W. G. Hillen W. H. Carrier	H. W. Worsham L. L. Lewis

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ANDERSON, CARROLL SIMMS, Mgr., American Blower Co., Los Angeles, Calif. (*Reinstatement.*)

ARMITAGE, JOHN BARNUM, Engr., Austin Co. of California, Los Angeles, Calif.

BARKER, CHARLES M., Office Mgr., The B. F. Sturtevant Co., Los Angeles, Calif.

BRADFIELD, PAUL EDGAR, Sales Engr., Crane Co., Los Angeles, Calif.

CAMPBELL, RAY W., Sales Engr., L. J. Mueller Furnace Co., Milwaukee, Wis.

CLANCY, GILBERT EUGENE, Gay Engineering Corp., Los Angeles, Calif.

DARLINGTON, ALLAN PALMER, American Blower Corp., Detroit, Mich.

ELLINGWOOD, ELLIOTT L., Consulting Engineer, H. W. Hellman Bldg., Los Angeles, Calif. (*Reinstatement*).

ENGLISH, HARROLD, President, English & Lauer, Inc., Los Angeles, Calif.

GILLE, HADAR, Partner, Hugo Theorells Ingeniorsbyra, Stockholm, Sweden.

GUNZEL, RUDOLPH MILTON, R. M. Gunzel & Co., Los Angeles, Calif.

HECHT, FRANK HENRY, B. F. Sturtevant Co., Pittsburgh, Pa.

HORTON, GEORGE H., Asst. Supt., Howe Bros., Los Angeles, Calif.

HUNGERFORD, LEO, Chief Engr., Fox West Coast Theatres, Los Angeles, Calif.

KIESLING, JUSTIN A., Geo. J. Robischung, Inc., Houston, Texas.

KINTZ, LESLIE H., Sales Engr., Johnson Service Co., Detroit, Mich.

KRATZ, ROBERT WAYNE, Paramount Famous Lasky Corp., Hollywood, Calif.

LAUER, HAROLD B., Vice-President, English & Lauer, Inc., Los Angeles, Calif.

MARQUET, CLYDE M., Vice-President, Pacific Engrg. Co., Los Angeles, Calif.

MARSCHALL, PETER J., E. Vernon Hill Co., Chicago, Ill. (*Advancement.*)

MCCORMACK, PETER JOSEPH, Estimator & Engineer, W. I. Bulkeley, Inc., Cambridge, Mass.

RIEGER, FRED R., Berg Engrg. Corp., Huntington Park, Los Angeles, Calif.

KILPATRICK, WILLIAM SQUIERS, Chief Engr., H. S. McClelland, Los Angeles, Calif. (*Reinstatement.*)

ROBINSON, HARRY C., Heating & Ventilating Engr., Worcester, Mass.

TILTZ, BERNARD E., Pres., Tiltz Engrg. Co., New York, N. Y.

WARD, ROBERT N., Engr., Tiltz Engrg. Co., New York, N. Y.

WARREN, HARRY L., Heating Engr., Southern California Gas Co., Los Angeles, Calif.

WEAVER, JOHN ROSWELL, O. W. Ott, Los Angeles, Calif.

ASSOCIATES

BREMSER, HARRY A., Sales Repres., Hoffman Specialty Co., Los Angeles, Calif.

ESSER, FRED WILLIAM, Refrigeration Div., Marine Engineering & Supply Co., Wilmington, Calif.

GREENE, ALFRED REYNOLDS, Heating Contractor, Binghamton, N. Y.

HILLIARD, FORREST H., Sales Engr., U. S. Radiator Corp., Los Angeles, Calif.

JOHNSON, ARTHUR R., Crane Co., Los Angeles, Calif.

KEELING, HARRY BEAUFORD, American Blower Corp., Los Angeles, Calif.

KNECHT, CHARLES H., Mgr., Harry Knecht Co., Inc., West Collingswood, N. J.

LAMONTAGNE, JOHN M., Sales Engr., Westinghouse Elec. & Mfg. Co., Los Angeles, Calif.

LEWIS, MILTON, Sales Engr., Southwestern Engrg. Co., Los Angeles, Calif.

MCCLELLAND, HARVEY SLOAN, Htg. & Vtg. Contr., Los Angeles, Calif.

MILLIKEN, VINCENT DEWEY, Skidmore Corp., Chicago, Ill.

MUELLER, PAUL G., Secy. & Treas., Pacific Engrg. Co., Los Angeles, Calif.

NELSON, J. FRED, Contract Mgr. & Estimator, Southern Asbestos & Magnesia Corp., Los Angeles, Calif.

OREAR, ANDREW GIBBS, Pacific Coast District, Ilg Elec. Ventilating Co., Los Angeles, Calif.

RICH, CALVIN GEORGE, Branch Manager, Berg Engrg. Corp., San Diego, Calif.

SCHOURUP, WALTER EDWARD, L. J. Mueller Furnace Co., Milwaukee, Wis.

TABY, JOSEPH CONRAD, Owner of Business, Second and Walnut Sts., Shamokin, Pa.

UPDEGRAFF, LEE, Sales Engr., 405 So. Hill St., Los Angeles, Calif.

WAID, GLEN HAROLD, Scott Valve Mfg. Co., Detroit, Mich.

WARREN, FREDERICK ANTON, Salesman, U. S. Radiator Corp., Atlanta, Ga.

JUNIORS

BINFORD, WILMER MERRIMAN, Contract Mgr., Southern Asbestos & Magnesia Corp., Los Angeles, Calif.

BOUEY, ANGUS J., Sales Engr., B. F. Sturtevant Co., Los Angeles, Calif.

BULLEIT, CHARLES R., Plumbers Supply Co., Evansville, Ind.

EARHART, JOSEPH STANLEY, Sales Engr., English & Lauer, Inc., Los Angeles, Calif.

KENDALL, EDWIN H., Sales Engr., English & Lauer, Inc., Los Angeles, Calif.

LAWLER, MATTHEW M., Chief Engr., Cooling and Air Conditioning Corp., Los Angeles, Calif.

Heating · Piping and Air Conditioning EDITORIALS

Pooling of Information

As a people, we of America are apt to be a little bit boastful of the standing of this country as a manufacturing and producing unit. This is a most natural pride for any loyal citizen but it is one that can be justified by an analysis of the facts. American manufacturing processes have arrived at the present high point largely because of the fine cooperative spirit among all parties concerned. The spirit of "Whatever benefits the whole, benefits me" is more prevalent in this country than in any other country in the world, for which we should be very thankful.

Notice especially the tone and technique of most of the papers presented before the technical societies. There is a freedom in presenting and discussing details relating to the most intimate mechanical and scientific processes that is positively surprising.

One reason for this may be the cordial relationships existing between the national scientific and technical societies on the one hand and the leaders of many of our great production plants on the other. These men frequently hold leading positions in and help to guide the destiny of these same technical societies, along with the most scientific and technical men the country affords. In so doing, they are not only disseminating valuable knowledge and profiting the community at large, but they themselves receive corresponding inspiration and knowledge to take back to their respective production centers.

Another reason for this free interchange of knowledge may be attributed to the influence of the technical college. More and more, the technical graduate is getting into industry, and more and more the industries are looking to the technical schools to help them solve their problems. A spirit of mutual cooperation and benefit is increasingly noticeable.

It should be the interest and duty, then, of every individual in the group to advance the knowledge of the group. This fact is recognized by progressive engineering organizations and they make it a part of their business to reveal fundamental engineering information which they have acquired through research or their years of experience. This willingness to pool information has contributed greatly to the advancement of engineering and to manufacturing.

Simplification of Nomenclature

According to a news release recently distributed by the *Department of Commerce* a noticeable phase of the simplification program has been a decided quickening of interest on the part of the consumer. According to George A. Cooper of the Division of Simplified Practice of the Bureau of Standards "Where five years ago nearly all simplified practice recommendations were initiated by manufacturers and brought to consumers and distributors for approval or modification, there are simplification programs developed today at the instance of purchasers."

Elsewhere in his paper "Recent Economic Effects of Simplification," Mr. Cooper stresses the need of standard codes in cataloging mechanical equipment which will result, he says, in economy for the purchaser and the producer. The variety of names and numbers to designate pieces of equipment in the scores of catalogs in the engineer's office is heralded as an inefficiency which should be corrected.

It is understood that the *American Society of Heating and Ventilating Engineers* contemplates a committee on investigating the nomenclature of the heating and air conditioning field. Several suggested simplifications are made in an article on that subject in this issue.

Simplification and standardization have proved their value in many fields; it is hoped the support of heating and air conditioning engineers will be lent to any revisions or simplifications of nomenclature it seems advisable to make.

Tell the Manufacturer

It is common these days for large manufacturers to have their own testing laboratories. But the greatest testing laboratories in the world are the factories, schools, hospitals, apartments and other buildings in which mechanical equipment is used day after day and year after year.

No manufacturer is satisfied with a laboratory

test only. The device or equipment must be tested in actual operation on the job before he can know that it will function satisfactorily.

Here is where the maintenance engineer can be of great service to the manufacturer and to the advance of engineering as well. When a person can perform a service that will benefit a large number of people and can do it with little effort, we submit that it then takes on the form of an obligation.

It is strongly recommended that maintenance and operating engineers report to manufacturers the manner in which their equipment functions. Too often the maintenance engineer will tinker with a piece of equipment, blame the manufacturers, and keep his dissatisfaction to himself. There is no manufacturer these days who would not welcome the opportunity to discover what the trouble is and make his equipment function as it should. Equipment is not sold these days merely to make one sale. The manufacturer who does not recognize this quickly goes out of business.

The tendency is toward keeping complete performance data and this information is not only the essence of an efficient maintenance department but is of vital importance to the manufacturer as well.

One maintenance engineer, commenting on the above, says "I think the solution is for the manufacturers to stress the advantages of their sales forces checking upon old customers and past sales with the same energy they use in getting new business."

Heat Waves and the Future

The heat wave which has afflicted the major portion of the United States has not been without its benefits; it has at least served to bring to the attention of a large part of the public that the air conditioning engineer is prepared to make the home as comfortable atmospherically as the theaters, the hotels, and other public and semi-public buildings.

The newspapers of Chicago (in the center of the heat-stricken area) have devoted considerable space to the advantages of air conditioning and to predicting that air conditioning for homes will soon be as necessary as is heating. McCutcheon, famed cartoonist of the Chicago "Tribune," allotted his usual space on the front page of that journal to the query "Why not have some system for artificially cooling homes as well as heating them?" An editorial on the subject was also published, which evidently caused considerable comment, for a few days later, in another editorial, the "Tribune" said: "We have been much impressed by the prompt reaction to our editorial of a few days ago on air cooling as the next step in the march of comfort. We have had a number of letters from firms and individuals" (one of them being Heating, Piping and Air Conditioning) "advising us that devices now exist for air cooling and that they are not too expensive for pretty general use. If this is true, as we hope it is, what seems to be lacking is intelligent and energetic exploitation of a great opportunity, a real want; in other words sound and effective salesmanship."

The St. Louis "Post-Dispatch" did some of the salesmanship necessary before the general public is thoroughly informed as to what air conditioning can do when it ran, on five consecutive Sundays, a full page under the heading "Heat when it's cold—cool when it's hot!" Each week there appeared an article by a nationally known authority on the subject, and the feature was received with considerable interest.

A Chicago newspaper, commenting on the new air conditioned dining car which made its first run on the "Chief" of the Santa Fe road the first part of August, said, editorially, ". . . air cooling, we predict, will soon be used on all passenger trains, and we do not except suburban. On the contrary, there is real need for it on suburban cars. There is no trial of the commuter so unpleasant as the intense heat during hot spells of the suburban cars, constructed of steel and heated to oven temperature by hours under the sun in the railroad yards."

The "Journal of Commerce" (daily of Chicago) says that "The American people have grown accustomed to a prodigious advance in their ways of living . . . they are ready for the next step. This is the mechanical refrigeration of the indoor atmosphere." (It is, and may we add the proper humidification of the indoor atmosphere during the winter months?) Further on, the "Journal" says, "The eyes that have been searching the horizon for a great industry that will stimulate business . . . may well rest content on the food-and-air refrigerator industry. The potential market for this is practically nation-wide; and, besides, there is an incalculable foreign market. . . . It is agreeable to picture the American householder sitting in midsummer in a cooled home. . . ."

As the growth of air conditioning is so dependent on increased general knowledge of what it can do, it is gratifying to see considerable attention paid to it in the general press. The trade and class publications are also showing increased interest in the activities of the air conditioning engineer; numerous requests have been received by Heating, Piping and Air Conditioning asking permission to reprint articles from this publication. These requests have, of course, been granted as Heating, Piping and Air Conditioning is firmly convinced of the necessity of widespread knowledge about air conditioning and the numerous advantages it promises to everyone.

Truly, the practically universal use of air conditioning *is* the next step in human comfort. And, while such widespread use is inevitable in time, it can be hastened by the publication of information on its advantages in the general press and in journals as close to the field as is Heating, Piping and Air Conditioning.

PRACTICAL PIPING PROBLEMS

Keeping Gasket Joints in Condition
Facing Flanged Joint of Screwed Type
Repairing a Compressed Air Line

by
W. H. WILSON

Replacing Gaskets

Replacing gaskets in flanged pipe joints can be reduced by careful initial installation and the selection of proper gaskets. Gaskets are often of the ring type, to fit inside of the flange bolts, which allows the pressure exerted by tightening the bolts to be distributed over a smaller gasket face area and consequently it is more effective. For pressures in excess of 125 lb. the gasket contact faces are usually raised slightly from the rest of the flange. For suction lines, however, some pipe men prefer a full-face gasket.

The recessed type of joint, with male and female contact faces, has an advantage for high pressure work, as it holds the gasket securely in place.

Phonographic finish, which, by the way, was used many years ago, is used in many cases and affords a good gripping surface for the gasket.

Cleanliness of Faces Important

Gasket contact faces on pipe flanges, valves and fittings should be scraped clean before the gasket is applied. This can be done when the pipe is laying on the floor, before installation. Cleaning the faces is difficult when replacing gaskets in existing pipe lines, as there may be very little room to separate the flanges in order to allow working space to scrape the gasket faces. This is made all the more difficult if the joint has male and female contact faces. The shut-off valve may be leaking and in the case of a steam line the hot vapor may be coming out of the top of the joint and hot water from the lower part.

In such cases there is a great temptation to do very little cleaning on the gasket faces, drop the gasket in place, tighten the bolts that can be conveniently reached, tighten the ones in the close places less than the others and trust that the joint will not leak. Gaskets installed in this manner are usually unsatisfactory.

The escaping steam vapors can be kept out of the way by using a hose to blow compressed air on the joint, or an electric fan can be used for the purpose. Opening a valve to the atmosphere or an exhaust back pressure valve to cause a draft through the pipe line is helpful in carrying the hot vapor away. A temporary connection to a vacuum line can be used to advantage in many cases.

Gaskets of sheet asbestos and similar material are likely to curl when being inserted in the joint if steam vapor is escaping. Under these conditions the gaskets are hard to land in their proper place. This difficulty can be overcome by inserting two thin sheets of steel or galvanized iron with the gasket between and then withdrawing the steel sheets, leaving the gasket in place.

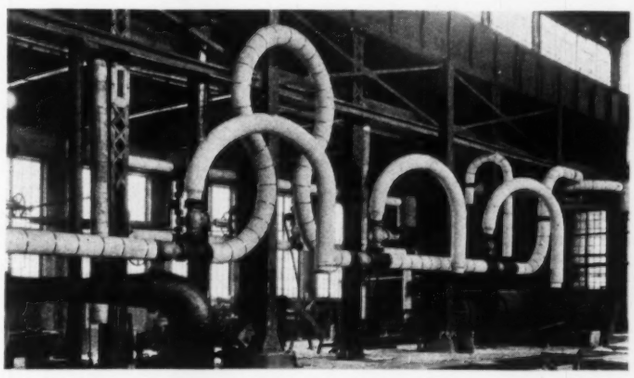
Tightening Bolts

Gaskets should be inserted carefully and the flange bolts tightened evenly. This is self evident to anyone, yet as simple as it seems it is the cause of many gasket joints leaking after installation. Flange bolts should be tightened evenly and carefully and every bolt pulled up tight, but the bolts should not be overstressed.

Bolts with easy running nuts should be put in the bolt holes that are the most difficult to reach with the wrenches and these bolts tightened first, when working on flanged joints in close quarters. Tightening the bolts easy to reach, last, will help, to some extent, to increase the tightness of the bolts in the places that are hard to reach with the wrenches. Stud bolts with a nut on each end of the bolt are time-savers for joints where the working room is limited. It should be understood that room should be provided for flanged joints, so that it would not be necessary to tighten joints, in this manner, as the proper way is to tighten the bolts evenly, but many piping installations will show flanged joints in places that are almost inaccessible.

Piping Must Be Installed Properly

The gaskets being in place and the joints completed, the care and maintenance begins. Gasket faces must be true to make a tight joint and must be kept in this position, if the joint is to stay tight. Good and sub-



stantial pipe hangers and supports should be used. The weight of the pipe line should be carried by the hangers and supports, so that there will be no strains on the joint. Vibration in pipe lines frequently will cause a joint to leak. Tie rods and braces should be installed, where needed, to correct this trouble,

In order that these braces do not affect the expansion travel in a pipe line, it may be necessary to provide tension springs in connection with the braces.

Temperature changes are detrimental to pipe joints of all kinds. Joints subject to temperature changes should be well insulated.

Condensation in steam lines causes trouble from gaskets leaking and should be removed by properly located drains and traps.

Gaskets for oil lines should be of a material that is not affected by the oil that is transmitted by the line.

Trouble is usually experienced with flanged joints in a pipe line that supplies steam intermittently, such as a steam line that is shut down at night during the summer season and turned on during the day.

Gasket renewals on pipe lines of this type can be reduced by going over the joints periodically, tightening the bolts if necessary. When steam vapor starts to issue from the joints, the bolts should be removed one at a time, and new ones used or a die run over the threads of the old ones, until the nuts move freely by hand. The bolts should then be replaced in the joint and pulled up tight. By this method many bad steam leaks can be stopped and gasket joints kept in service.

Another method of repairing leaking gasket joints that is widely used is the forcing of special compounds into leaking joints with equipment designed for the purpose. The steam pressure is not shut off while this work is being done and expensive delays and shut downs are avoided. Many exceptionally bad leaks can be closed up in this manner.

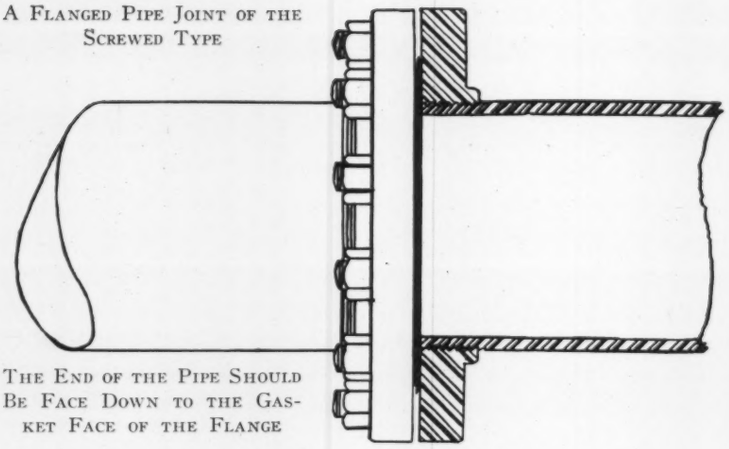
Facing Flanged Joint of Screwed Type

The drawing shows a flanged pipe joint of the screwed type where the flange is made up on the pipe until the end of the pipe is flush with the gasket contact face of the flange.

To insure a good joint and a true gasket face it is necessary to screw the flange on the pipe until it makes up tight and projects beyond the gasket face on the flange and then face the end of the pipe down to the gasket face of the flange. This will give a true gasket face from the inside of the pipe to the outside of the gasket contact face on the flange, which is the proper bearing surface for the gasket.

This means more machine work, but for pipe lines that are subject to a certain amount of vibration, temperature changes and the general wear and tear incident to industrial piping, it is good practice to do this extra work.

A Flanged Pipe Joint of the Screwed Type



The End of the Pipe Should Be Face Down to the Gasket Face of the Flange

Pipe Repairs

Occasionally, operating conditions require that pipe repair work be done during the regular working hours without interfering with the production work of the plant.

The diagram below represents the location of a section of compressed air piping where is was necessary to renew a gasket in a flanged joint and at the same time furnish a required amount of compressed air to a shop located some distance away.

A pressure of 100 lb. was maintained on the pipe lines at all times with the exception that during a short period a lower pressure was sufficient for the work that was being done at that time.

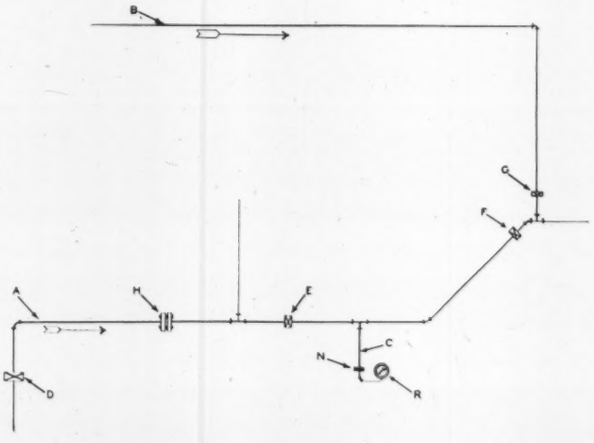
Referring to the diagram, *A* represents a 4-in. line, the main supply to the distant shop; *B*, a secondary, 1½ in. supply line connecting with the main supply. *D*, *E*, *F*, and *G*, are shut-off valves and *C*, a drip pocket for entrained moisture, with a drain valve at *N*. *H* is a 4-in. flanged joint, leaking, which required the renewing of the gasket. This joint was located high above the floor, in the roof trusses.

An additional difficulty was the small amount of clearance in the line to enable the flanges to be separated in order to scrape the faces clean and insert the new gasket.

Procedure in Repairing

The first step in the repair work was to place the ladders in position for working on the flanged joint. Next the pressure gage, *R*, was applied to the drain valve *N*, and this valve opened. Then valves *D* and *E* were closed.

Repairing a Pipe Line Without Shutting Off the Air Supply



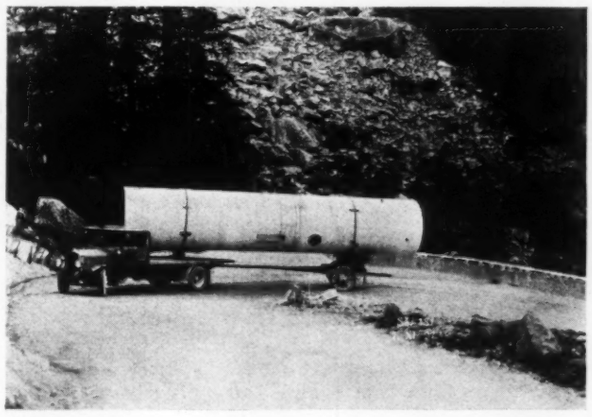
The leaking joint reduced the pressure in this section of the piping and it was possible to remove the flange bolts one at a time, oil and replace them so that the bolts could be handled quickly when the gasket was to be replaced. During this time a close watch was kept on the pressure gage and as soon as possible valves *D* and *E* were opened.

Arrangements were then made to force the pipe line apart when the flanges were to be separated to provide room for scraping the flanges and inserting the new gasket. This was accomplished by using a small jack screw and some wedges, farther back on the line where it passed through the building wall.

When the operating conditions permitted sufficient air to be supplied through the secondary supply line, *B*, tools and gasket were on hand, valves *D* and *E* were closed. The flanged joint was separated and the new gasket put in place. The flange bolts were then tightened, valves opened, tools and equipment removed. The work was completed without any interruption to the compressed air service to the shop.

This arrangement can be used in many similar cases. Frequently a temporary connection can be made by using hose or pipe as an auxiliary supply.

By using a pressure gage or gages at certain points it will enable the workmen to keep a close check on important pipe lines that must still furnish a continuous supply while the repair work is in progress. A seemingly small, yet important, matter in work of this kind is to have all of the tools needed on hand before commencing the work and to have the bolts and nuts in good condition so that no time is lost when removing and replacing them.



THE CONSTRUCTION OF A LARGE HYDRO-ELECTRIC POWER PLANT IN SWITZERLAND PRESENTED SOME DIFFICULTIES TO THE TRANSPORTATION OF MATERIALS, AS MAY BE SEEN FROM THE ABOVE VIEW

Air Conditioning and Furniture

The National Lumber Manufacturers' Association has its headquarters in Washington, D. C. This association encounters many engineering problems and for this reason employs Frank P. Cartwright as chief engineer. In discussing some of the engineering problems of the industry, Mr. Cartwright recently told the following story:

"A friend of mine, who has occupied for years a high position in the Government foreign service, has been a very enthusiastic collector of rare and beautiful examples of old furniture. For the most part his acquisitions, which in some instances are of the museum character, have been picked up in Continental homes and castles, and the writer's own reflection has been that the original owners in dispensing with them must have valued comfort more than antiquity. However this may be, the official in question collected and kept in a Washington warehouse (not kept at living room temperature in the heating season) a rare and impressive store of house furnishings. Some two years ago a suitable residence was built to accommodate these treasures.

"Unfortunately, the collector neglected to consider his heating and ventilation. Soaked and seasoned as they were by centuries of stone-walled dampness and heatless rooms, the antique pieces gave up their store of moisture to the insatiable thirst of interior winter humidities. Ominous creaks and loosening joints appeared. Shrinking members exposed fresh colored wood long hidden from sight. Cabinet doors swung silently open and stayed that way.

"Fortunately, the collector realized his difficulty and took prompt and effective measures to avoid future trouble by putting in a small air conditioning plant, which not only preserves his furniture from damage, but makes living conditions in the house ideal during the winter time.

"The moral has some application for any American home, though our modern furniture is made of more carefully seasoned material, which straddles the fence between summer humidity and winter heating conditions. A discerning builder, however, will find it well worth while to provide his residence with equipment capable of maintaining winter humidities above 40 per cent."

Conventions and Expositions

National Association of Power Engineers: Annual convention and exhibition, Sept. 8-13; Cleveland Auditorium, Cleveland, Ohio. Secretary, Fred W. Raven, 417 S. Dearborn St., Chicago.

American Welding Society: Fall meeting, Sept. 22-26; Congress Hotel, Chicago. Secretary, M. M. Kelly, 33 W. 39th st., New York City.

National Metal Congress and Exposition: Sept. 22-26, Stevens Hotel, Chicago.

American Society of Mechanical Engineers: Fall meeting, Oct. 13-17, French Lick, Ind. Annual meeting, Dec. 1-5, New York City. Secretary, Calvin Rice, 33 W. 39th St., New York City.

American Gas Association: Annual convention and exhibition, Oct. 13-17; Atlantic City Auditorium, Atlantic City, N. J.

Stoker Manufacturers Association: Nov. 11-13, Greenbrier Hotel, White Sulphur Springs, West Virgina.

National Association of Practical Refrigerating Engineers: Annual convention and exhibition, Nov. 11-14; Hotel Peabody, Memphis, Tenn. Secretary, E. H. Fox, 5707 W. Lake St., Chicago.

National Power Show: Dec. 1-6; Grand Central Palace, New York City. Manager, Charles F. Roth, Grand Central Palace, New York City.